EFFECTIVE CONTAMINATION CONTROL IN FLUID POWER SYSTEMS

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Acknowledgement –

This publication is adapted from a text researched and written by John Spencer, Distributor Sales Mgr.-United Kingdom, and Clifford Balmer, Engineering Mgr.-France, Vickers Systems Ltd., Europe.

Fig. 3: Critical clearances in a vane pump.

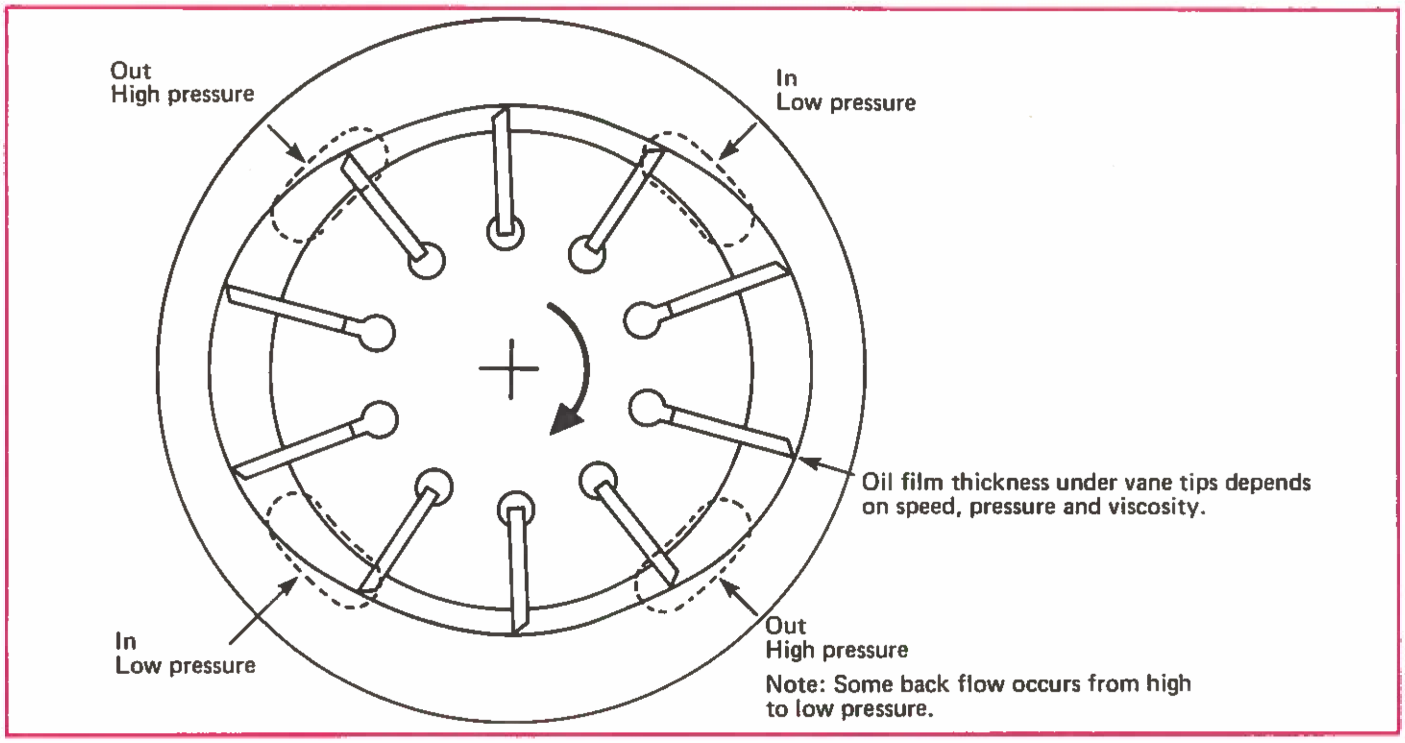
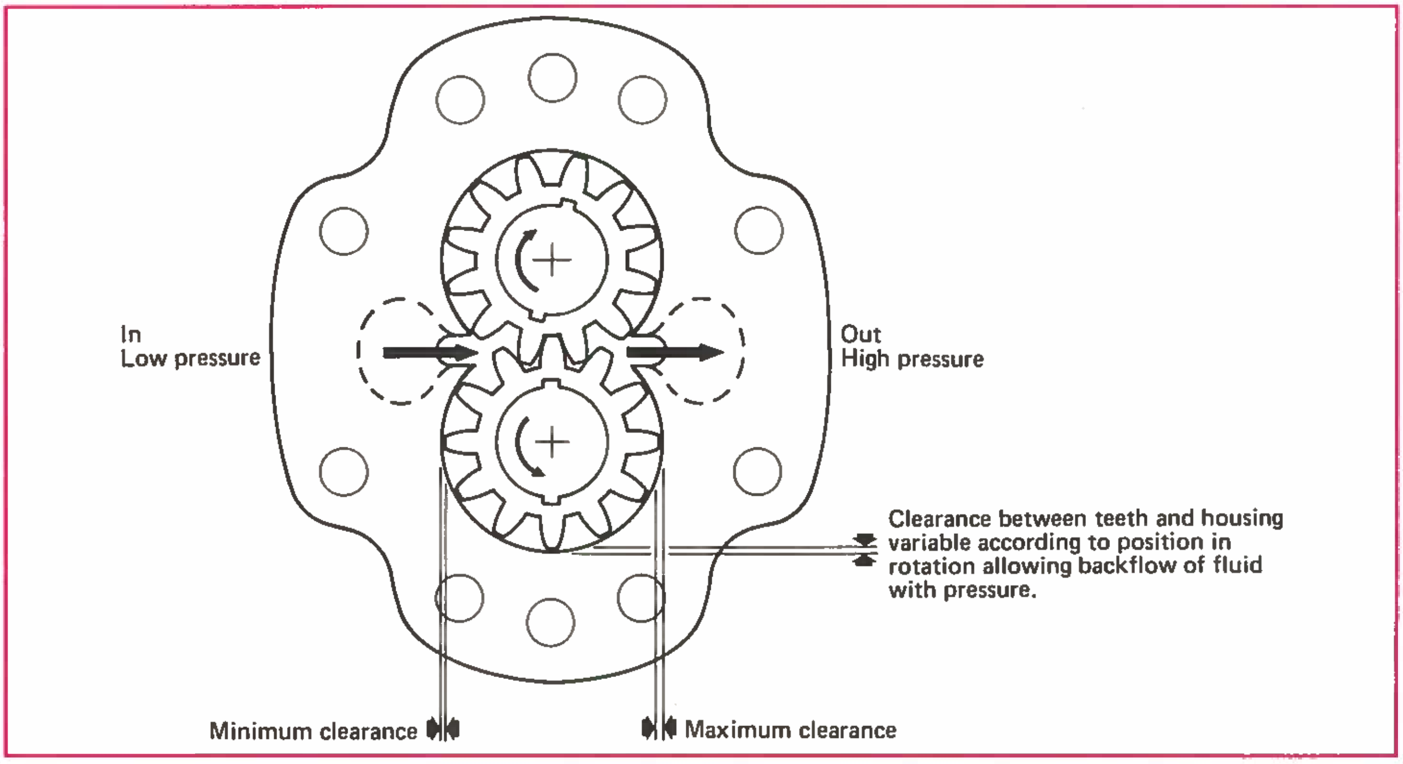

Out
High pressure
In
Low pressure
Oil film thickness under vane tips depends on speed, pressure and viscosity.
In
Low pressure
Out
High pressure
Note: Some back flow occurs from high to low pressure.

Fig. 4: Critical clearances in a gear pump.

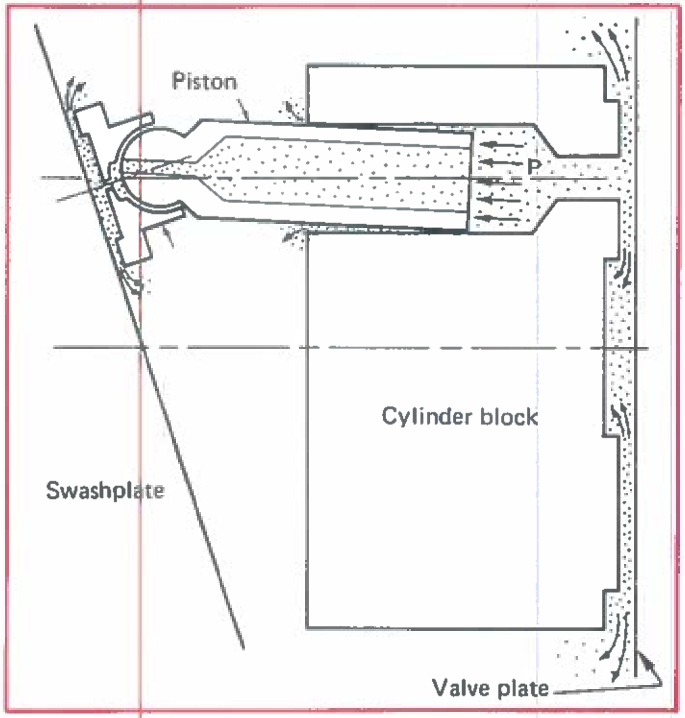

In
Low pressure
Out
High pressure
Clearance between teeth and housing variable according to position in rotation allowing backflow of fluid with pressure.
Minimum clearance
Maximum clearance

In many of the foregoing cases, the clearances are effectively self-adjusting under operating conditions, i.e. with increasing pressure clearances become smaller. Under adverse conditions, and particularly where there is shock loading, this increases the vulnerability to smaller contaminant particles. Even where clearances are nominally fixed, components under high loads may take up eccentric positions which again makes them vulnerable to smaller particles. It is extremely difficult to be precise about either the magnitude of these clearances, particularly under dynamic loading, or the effect of different size particles in the gaps.

However, from the data we do have and from field experience gained to date, we are able to suggest contamination levels which, if achieved, will result in an acceptable life for most pumps. These are presented in the next chapter, which deals with the specification of contamination levels.

The useful life of a pump should end when it no longer delivers the required output at a given shaft

Fig. 5: Critical clearances in an axial piston pump. Although piston clearance is nominally fixed, actual clearance varies with eccentricity due to load and viscosity.



speed, discharge pressure and fluid temperature. As a guide, 30% loss of flow has been used to indicate the end of the useful life. All too often, degradation goes undetected until finally catastrophic failure occurs, with vast quantities of contamination being released into the system. If, following such a failure, the system is not then properly cleaned, the life of the replacement pump will be reduced.

In the interest of the end user, the system designer should specify the minimum acceptable flow rate from the pump to achieve satisfactory machine performance. Means should be provided for monitoring pump output by inserting suitable instrumentation, either temporarily or permanently, so that routine checks can be carried out to reduce the risk of catastrophic failures. With piston units, it is usually a simple matter to measure case leakage, which can be a useful guide to pump condition.

Remember, to the end user, total costs are important; the failure of a low-cost pump may well result in expensive downtime. If, by the inclusion of a flow meter, such a failure can be avoided, the initial investment in a flow meter would be fully justified.

Motors

What has been written about pumps applies generally to motors of similar design.

It must be remembered that much of the contaminant passing through the pump may be reaching the motor also, where there will be a similar performance degradation. If, due to wear, the volumetric efficiency of the pump has fallen to 85% of its original value and the volumetric efficiency of the motor has fallen to, say, 90% of original, then the overall volumetric efficiency of the pump and motor will be down to 0.85 x 0.9 = 76.5% of the original value. For this reason contamination control is particularly important in hydrostatic transmissions to provide the necessary level of fluid cleanliness.

Directional Valves

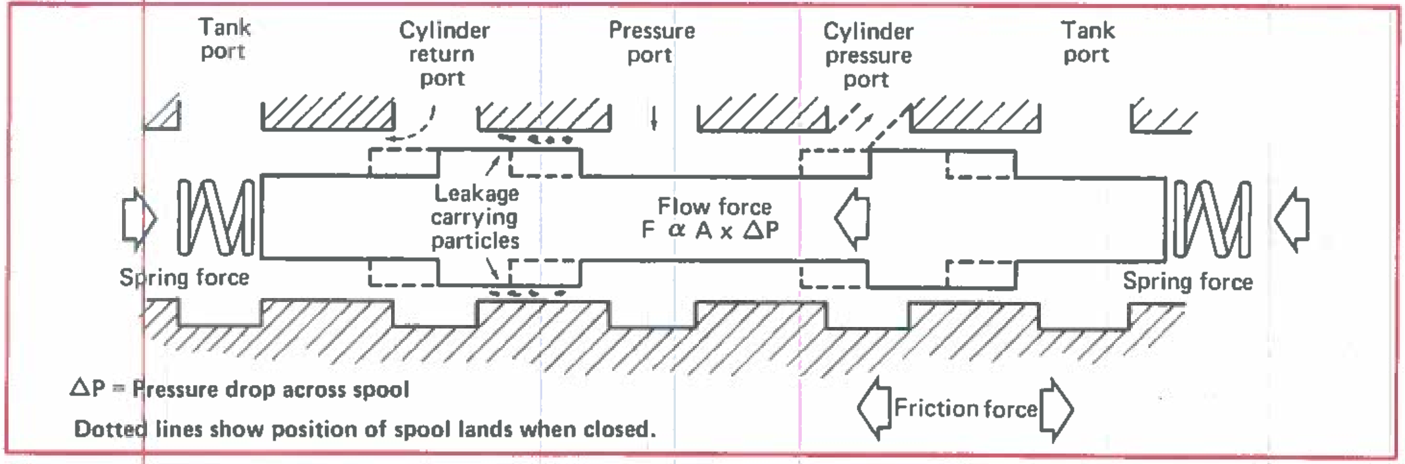
The radial clearance specified between bore and spool in most directional valves is in the range of 5 to 13 micrometres. As is well known, the production of perfectly round and straight bores is exceptionally difficult, so it is unlikely that any spool will lie exactly central in the clearance band. In a nominal 1/8 inch valve, a good spool is likely to have less than 2.5 micrometres clearance.

In an electrically operated valve, the forces acting on the solenoid are shown in Fig. 6. They are: flow forces, spring forces, friction forces and inertia forces.

Flow, spring and inertia forces are inherent factors, but friction forces are, to a great extent, dependent on filtration. If the system is heavily contaminated with particles similar in size to the radial and diametral clearances, higher forces will be needed to move the spool.

An even worse situation results from silting, where contaminant is forced into the clearances under pressure, eventually leading to breakdown of the oil film and spool hang-up (Fig. 7).

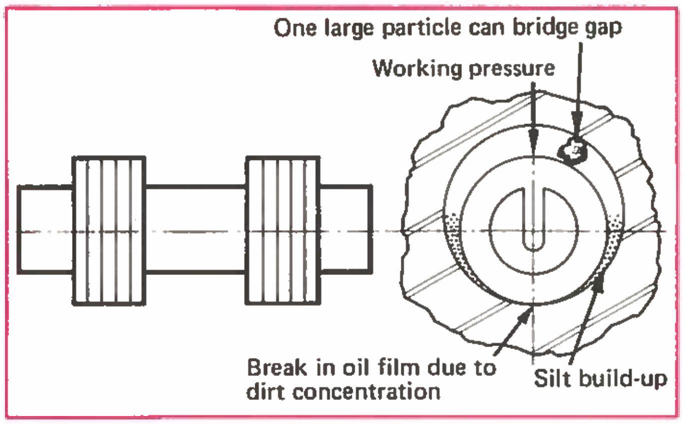
Fig. 6: Valve spool critical clearances (with flows and forces).



ΔP = Pressure drop across spool

Dotted lines show position of spool lands when closed.

Fig. 7: **Valve spool critical clearance. Eccentricity is the normal condition in many cases.**


One large particle can bridge gap
Working pressure
Break in oil film due to dirt concentration
Silt build-up

This situation occurs where valves subjected to continuous pressure are operated infrequently. Such valves should preferably have local filtration of a very high order in the adjacent pressure line, but due account should be taken of possible pressure surges generated during component operation. The use of high efficiency filters as a special protection for single units or groups of units can result in the need for a very high dirt capacity, if the general level of filtration in the system is much lower.

Some idea of the forces needed to break this spool hang-up, compared with the forces available from the solenoid, can be gained from the example of a nominal 1/8 in. valve operating at 3000 psi. If a valve of this type remains in the spring offset or energized position for a lengthy period of time, silting takes place between spool and bore to produce total immobility. The force needed to overcome this state has been found by experiment to be of the order of 30 pounds, but both spring and solenoid could exert only 10 pounds. Thus the effect of the silt causes total system failure.

Pressure Controls

Highly abrasive particles in high velocity streams of oil erode internal surfaces. This situation is common to pressure controls, particularly relief valves which are subjected to maximum system pressure drop and velocities of the order of 90 ft/sec. Pilot control stages generally see low volumes at high velocities, and heavy contamination affects both their stability and repeatability.

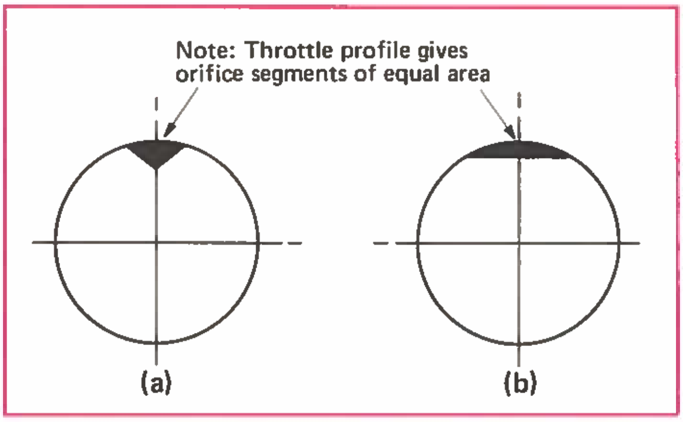
Flow Controls

The contamination tolerance of flow control valves will depend very much on the orifice configuration.

Fig. 8, for example, shows two orifices which are of entirely different shape although having equal areas. The groove type (a) will tolerate a high contamination level, except when used at a low setting, whereas type (b) is much more prone to silting at all settings.

With all types of pressure compensated flow controls, the performance of the pressure reducing element can be considerably affected by contamination, irrespective of valve setting. Damage to the metering orifice can also occur, which will become particularly apparent at lower settings.

Fig. 8: **Flow control valve throttle sections. Profile (b) is more prone to silting.**



Generally speaking, all spool-type control valves are affected by contamination in the system, especially at high pressures. The effects are likely to be magnified if precise axial positioning of the spool is necessary as, for example, in pressure reducing valves where limited forces are available to operate the spool. On the other hand, poppet valves, though affected by large particles of contamination, tend to be far more tolerant of silt due to the self-cleaning action of the seat. However, erosion is still likely to occur.

Summary

It can be seen from the foregoing that an individual large particle arriving at the wrong place at the wrong time can cause catastrophic failure.

Surfaces within components should be separated by an oil film, the thickness of which may be continually changing. When this gap is bridged by contaminants, wear will occur, thereby generating further particles which may be ground into many more smaller particles. Fine particles, individually or in small quantities, may not cause damage, but if present in larger concentrations may lead to failure through silting.

The objective must be to obtain the most economic combination of contamination control and contamination tolerance for a given reliable system life under known performance and environmental conditions.

CHAPTER 4

Specifying Contamination Levels

As stated previously, 25 micrometres is a typical general level of filtration, specified without regard to working pressure, local environment or duty cycle. We know from experience that, under similar environmental and operating conditions, the effectiveness of a 25 micrometre filter will vary depending on its location in the system. Also, filtration performance may actually improve under steady flow conditions as the pores tend to clog. Under varying or intermittent flow, however, the result can be very different because contaminant particles are dislodged from the pores, allowing more fine particles to pass through.

It is obvious that the conditions of use have the greatest effect on the contamination level obtained with a specific filter. Thus it is generally unwise to offer without qualification, or accept without question, a blanket recommendation of, for example, the use of a 25 micrometre filter. For the user it means risking system investment, and for the manufacturer it makes the administration of warranties extremely difficult.

What is required is an acceptable contamination level to be specified and achieved, which then becomes the end user's responsibility to maintain. The system designer, after paying due regard to the environment, system pressure and duty cycle, selects an element rating and the location for a filter or filters that will successfully control and maintain the desired level.

At present, this more systematic approach is rather difficult to implement fully because we cannot yet define, for example, what exactly is a 10 micrometre particle. There is also the difficulty of extracting information from some filter manufacturers as to what their product will actually achieve in practice. However, from our experience to date, we can at least make a start in the right direction.

Useful data has been obtained by:

1. Taking samples from systems in a wide range of industrial and mobile applications.
2. Analyzing field failures due to contamination.
3. Monitoring over a three year period the cleanliness level of a particular system, and at one year intervals checking the critical clearances of the pump for wear.

From 1 and 2 it can be shown that there are many general industrial applications performing satisfactorily where the total numbers of particles above 5 micrometre are in the 100,000 to 300,000 per 100 ml zone. But they are mostly in the lower pressure ranges or have units operating well within their rated capacity. When high performance units, however, are employed with 5 micrometre particle counts above 100,000 per 100 ml, the chances of experiencing trouble are high.

Monitoring (3) the performance of one particular system revealed that very little wear had taken place in the pump, but at no time during the 3 year period did the particle count above 5 micrometre exceed 16,000 per 100 ml. The same type of pump from a similar machine incorporating the same degree of filtration was recently dismantled after a 5 year period (estimated hours 40,000) and showed only slight wear.

From the above and the work of other investigators, we can produce the chart shown in Fig. 9. While this helps meet our needs, whether a system is critical or very important will depend largely on the type of components used and the system design pressure. For this reason the graph shown in Fig. 10 has been produced to give the recommended contamination level, knowing the system pressure and having analysed the dirt sensitivity of the system components. The contamination levels are specified per ISO Solid Contamination Code, hereafter referred to as "ISO S.C. CODE", an explanation of which is given in Chapter 8.

CHAPTER 5

Selecting the Filter

We now come to the vital question of the filter or filters needed to give the desired contamination level. To answer the question we must first examine the various ratings used by manufacturers.

Nominal Rating

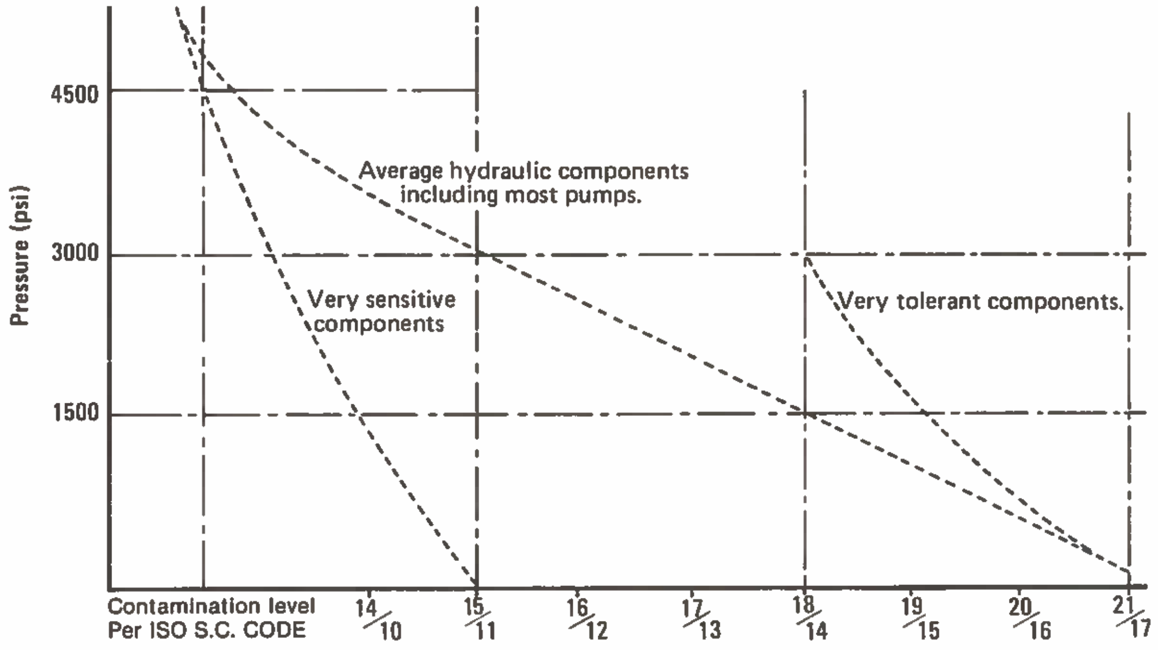
Specifications MIL-F5504A and MIL-F5504B were established for determining nominal ratings. Version A defines a 10 micrometre filter as being able to remove 98%, by weight, of all particles of the elected

Fig. 9: Suggested acceptable contamination levels for various hydraulic systems.

TARGET CONTAMINATION CLASS TO ISO S.C. CODE		SUGGESTED MAXIMUM PARTICLE LEVEL		SENSITIVITY	TYPE OF SYSTEM
5µm	15µm	5µm	15µm		
13	9	8,000	500	Super critical	Silt sensitive control system with very high reliability. Laboratory or aerospace.
15	11	32,000	2,000	Critical	High performance servo and high pressure long life systems, i.e. aircraft, machine tools, etc.
16	13	64,000	8,000	Very important	High quality reliable systems. General machine requirements.
18	14	250,000	16,000	Important	General machinery and mobile systems. Medium pressure, medium capacity.
19	15	500,000	32,000	Average	Low pressure heavy industrial systems, or applications where long life is not critical.
21	17	2,000,000	130,000	Main protection	Low pressure systems with large clearances.

Fig. 10: Suggested cleanliness level for good life.

Note: This graph assumes viscosity to be within recommended range.



Figures relate to cumulative particle concentrations above 5μm and 15μm respectively (see chapter 8).

contaminant (AC fine test dust) larger than 10 micrometres at a certain high concentration. Version B defines a 10 micrometre filter as being able to remove 95%, by weight, of 10-20 micrometre glass beads at a high concentration. Although little use has been made of these particular specifications, many manufacturers use similar tests to provide nominal ratings for their filters.

Such tests have two major limitations. First, they do not limit the maximum size of particle allowed to pass through the element, and from tests it has been found that filters meeting these requirements can pass particles up to 200 micrometres. Second, the high concentration of contaminant added is not typical of conditions experienced in a normal system. In practice, the particles approach the filter in small concentrations and those particles that are smaller than the mean pore size pass readily through the filter, as long as the filter medium remains reasonably clean.

For these reasons, there is a good case for discontinuing the use of nominal ratings.

Absolute Rating

The NFPA Fluid Power Glossary of Terms defines the absolute rating as being the diameter of the largest hard spherical particle that will pass through a filter under specified test conditions. This is an indication of the largest opening in the filter element.

This measures only one physical characteristic of a filter medium, i.e. the order of magnitude of the larger holes.

The Bubble Test

This is a test used by manufacturers to determine the area of greatest porosity. It is achieved by applying air pressure to the inside of the filter element, which is submerged in a liquid such as alcohol to wet the filter media. The operator rotates the filter element at each pressure level and records the pressure at which the first stream of bubbles emitted from the filter element. The test can be continued to measure the pressures of the second, third, fourth, etc. largest hole. By continuing to slowly increase the pressure, a point is reached called the 'open bubble point' at which air bubbles appear over the entire surface of the filter element. This is a simplified method of measuring the mean pore size.

It is claimed that precise results are not obtainable from the bubble test, which is unfortunate because such a simple test would be invaluable. As it is, its main use is in the quality control of elements to ensure there is no damage to the media or a bad seal.

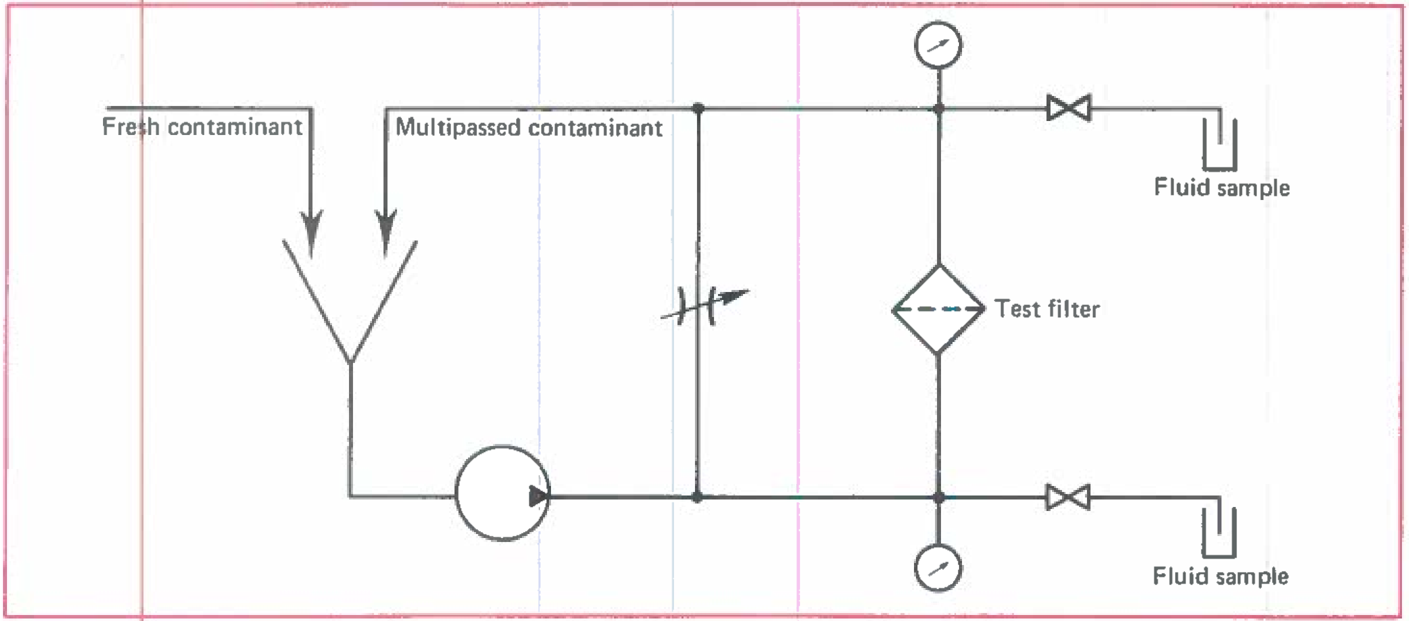
Mean Filtration Rating

This is a measurement of the average size of the pores of the filter media. This is a very significant rating, since it is a measure of the particle size above which the filter starts being effective. It can be measured using the 'open bubble point' method just described.

Multipass Filter Test

This test, designed to provide a means of describing the performance characteristics of a filter, involves the continuous injection of a controlled contaminant into a test system. As the contaminant can be removed only by the test filter, it will continue to circulate in the system unless captured. The separation capability of the test filter is monitored by analyzing upstream and downstream fluid samples. The dirt holding capacity is measured by the amount, in grams, of test contaminant which can be added to the system

Fig. 11: The multipass filter performance test.



before a specified terminal pressure drop across the filter is reached.

The mathematical relationship which describes the test is developed from the following expression:

$$\begin{matrix}\text{Number of}\\ \text{particles}\\ \text{downstream}\\ \text{of size} > x\mu\text{m}\end{matrix} = \begin{matrix}\text{Number of}\\ \text{particles}\\ \text{originally}\\ \text{of size} > x\mu\text{m}\end{matrix} + \begin{matrix}\text{Number of}\\ \text{particles}\\ \text{injected}\\ \text{of size} > x\mu\text{m}\end{matrix} - \begin{matrix}\text{Number of}\\ \text{particles}\\ \text{removed}\\ \text{of size} > x\mu\text{m}\end{matrix}$$

Beta Ratio. The separation characteristics are given by the Beta ratio, which is defined as follows:

$$\beta x = \frac{\text{Number of upstream particles larger than } x\ \mu\text{m}}{\text{Number of downstream particles larger than } x\ \mu\text{m}}$$

A Beta ratio of 1 indicates that no particle contamination is removed. A figure of less than 1 is clearly impossible unless the filter is unloading contaminants.

For a filter exhibiting a Beta ratio greater than 1, the downstream concentration of particles above a given size will stabilize to give an almost constant contamination level.

Practical Classification of Filter Performance

Whatever format is used by the filter manufacturer to give product performance information, the degree of filtration provided will basically fall into one of three categories depending on the degree of silt control. Typical data corresponding to these categories are given in Fig. 12, although the manufacturer should state the proper product classification. At present there is no universally recognized standard classification, but work being carried out by various bodies should eventually lead to an internationally agreed-upon definition.

Two adverse factors affect the actual performance of filters in service, namely pulsating flow and the sometimes uncertain performance of internal seals and bypass valves.

Pulsating Flow forces fine particles through the media, which would otherwise lodge among the fibres and between larger particles already intercepted. The effect may be compared with a sieve holding a mixture of stones, some larger and some smaller than the openings in the mesh; when the sieve is stationary, many of the small stones are retained but fall through when the sieve is shaken. Pulsating flow therefore increases the proportion of silt particles in the system downstream of the filter, and this is reflected in the performance curve of the filter (Fig. 13).

Bypassing by internal or external valving is acceptable for many systems since, in a number of passes, all the fluid eventually goes through the filter.

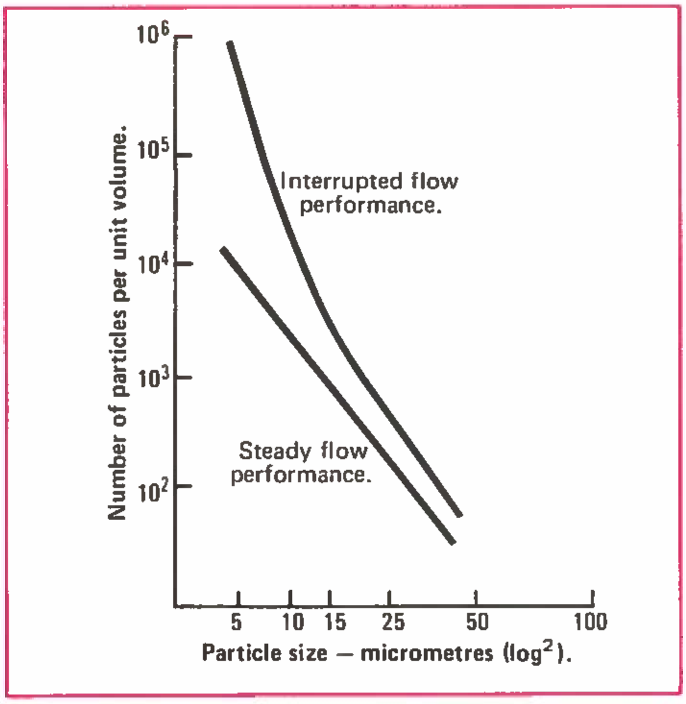
For filters other than those designed for permanent bypassing, it would normally be expected that the bypass be operating only when the element nears the end of its useful life. Therefore, the effects of premature opening of the bypass valve or a faulty internal seal need to be considered. Since bypassing, whatever the cause, does not discriminate between fine and coarse particles, the effect is to weight the contamination profile heavily at the coarse end. Fig. 14 by Dr. Fitch shows the effect of increasing percentages of bypass

Fig. 12: Definition of practical classification categories.

CATEGORY	NOMINAL RATING*μm	ABSOLUTE RATING
SILT CONTROL	½ to 1	3 to 5
PARTIAL SILT CONTROL	3 to 5	10 to 15
NO SILT CONTROL (Chip removal)	10 to 15	25 to 40

*Nominal ratings given for reference only. Their use is not recommended and this table should not be used for nominal/absolute conversion.

Fig. 13: Filter performance deteriorates under intermittent or pulsating flow. The effect is more marked with finer particles, which are forced through the media.



flows; note that even at 0.1% bypass the maximum particle size has almost doubled. At 1% it has more than tripled and at 10% it has increased by a factor of five.

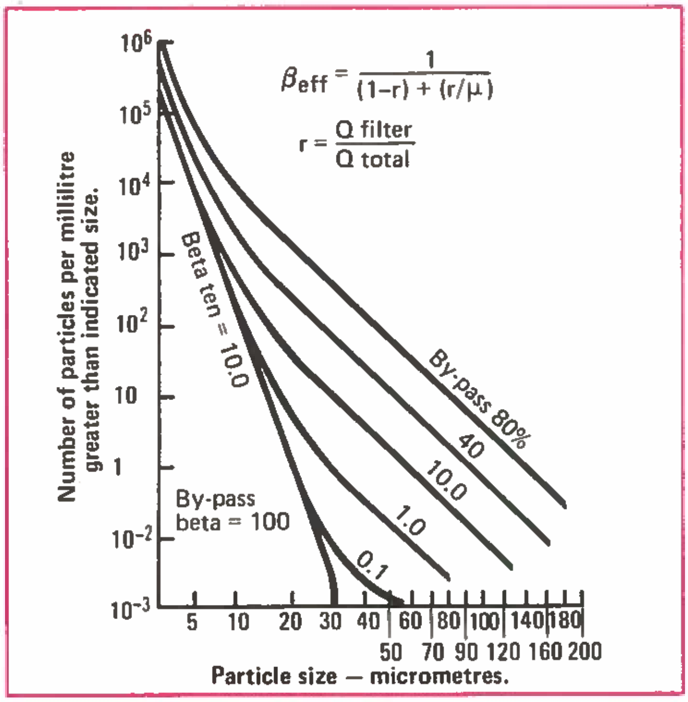
The practical significance of bypassing, and the need to provide for it, depend on the location of the filter in the system. For example, on inlet filters a bypass valve is mandatory in order to protect the pump from cavitation as the filter becomes blocked.

Pressure filters are intended to protect the system if there is catastrophic failure of the pump. If such a failure occurs when the filter is bypassing, then the protection is nonexistent. There is obviously a case for fitting non-bypass pressure filters where the need for this type is established. An element must be fitted that will withstand the total system pressure, which increases the cost of the filter. Clogging of the element causes system performance to fall off and a reliable indicator is essential to give early warning of this.

Where a bypass is judged to be permissible, the system designer must carefully weigh the effects of premature opening due to cold starts and surge flows. The elimination of surges, even those at low pressure, will reduce the tendency for the bypass to open under normal operation.

At present there is little information available on the extent of the adverse effect that pulsating flow has on filter performance. Hopefully, Beta ratios will eventually be quoted which relate to conditions line filters are subjected to in actual practice. Naturally, where steady flow exists, relating more closely to laboratory test conditions, more accurate predictions of filter performance can be made. However, using the limited knowledge at our disposal, likely performance levels for each of our categories have been produced in Fig. 15.

Fig. 14: Influence of by-pass leakage on filter performance.



On each graph a cumulative count of both 5 and 15 micrometre size particles has been shown. The reason for using these figures is because of the standard set by the ISO S.C. CODE explained in Chapter 8. For ease in relating to this document the range numbers are also quoted. In addition to fixed figures at the two values, the lines are extended to show expected trends for typical filters. Assuming these graphs relate to performance of the filter supplied by the particular manufacturer chosen, it is necessary to show that the actual requirement, when plotted, stays below the line drawn for steady or pulsating flow.

As an example, if we have a system with a working pressure of 2200 psi using components with average dirt tolerance, our cleanliness level graph (Fig. 10) tells us that the desired contamination level should be somewhere between 15/11 and 18/14.

Referring now to Fig. 15a and b, we see that under steady flow conditions this could be achieved by our partial silt control filter, but if pulsating flow is present a silt removal filter is required.

It can be assumed that most pressure and return line filters are subject to pulsating flow conditions. The reasons are given in greater detail in the chapter on filter location. On the other hand, off-line systems have the benefit of a steady flow rate condition.

Filter Sizing

It is current practice for manufacturers to state a flow rating at a specific clean pressure drop (see Fig. 16). While this is a guide to capacity, it may be necessary for the system designer to provide additional dirt capacity to ensure that the end user obtains an acceptable element life. All too often, filters sized purely on flow rate have a short element life. In choosing larger filter capacities greater initial expense

Fig. 15A: Assumed mean pore rating 3 micrometre. Curves show deterioration of filter performance with respect to particles below this size under pulsating flow.

SILT REMOVER

SIZE µm	CUMULATIVE COUNT	
	STEADY FLOW	PULSATING FLOW
5	16,000 (14)	64,000 (16)
15	1,000 (10)	2,000 (11)

Range number per ISO S.C. CODE shown in parentheses.

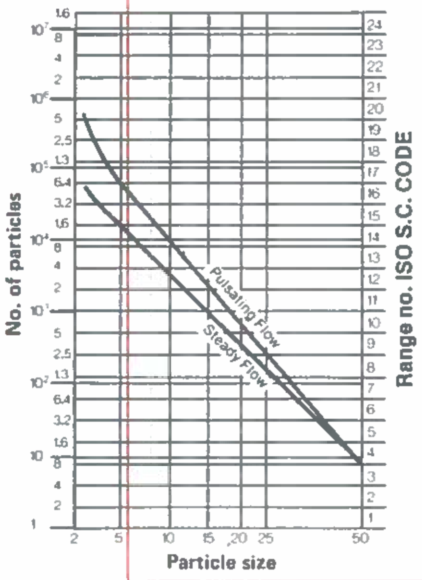


Fig. 15B: Assumed mean pore rating 7 micrometre. Curves show deterioration of filter performance with respect to particles below this size under pulsating flow.

PARTIAL SILT REMOVER

SIZE µm	CUMULATIVE COUNT	
	STEADY FLOW	PULSATING FLOW
5	32,000 (15)	25,000 (18)
15	2,000 (11)	4,000 (12)

Range number per ISO S.C. CODE shown in parentheses.

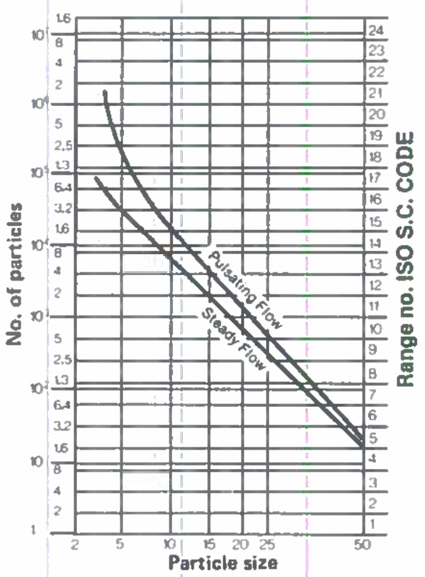


Fig. 15C: Assumed mean pore rating 15 micrometre. Curves show deterioration of filter performance with respect to particles below this size under pulsating flow. There is little or no control of 5 micrometre particles.

CHIP REMOVER

SIZE µm	CUMULATIVE COUNT	
	STEADY FLOW	PULSATING FLOW
5	250,000 (18)	16.000,000 (24)
15	16,000 (14)	64,000 (16)

Range number per ISO S.C. CODE shown in parentheses.

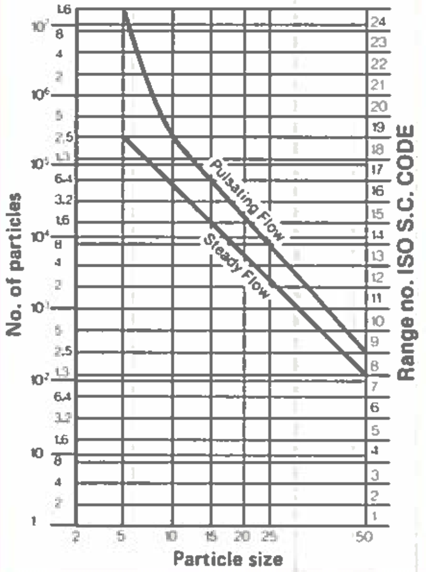
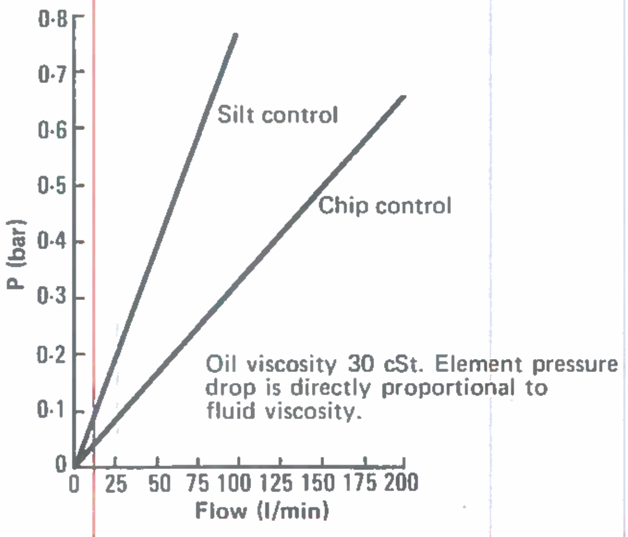


Fig. 16: Typical pressure drops for clean silt control and chip control elements of similar size.



Oil viscosity 30 cSt. Element pressure drop is directly proportional to fluid viscosity.

may be incurred, but this is almost certain to be recovered in reduced running costs, i.e. fewer element changes, reduced labor costs and less downtime.

Correct filter sizing requires relating the dirt entering the filter to the effective element area and the maximum allowable pressure drop. The relationship of area to pressure drop is not simple, however, and filter inlet dirt levels are rarely known.

There is a laboratory test (the 'comparative life' or 'dirt capacity' test) which is designed to compare the dirt holding capacities of hydraulic filters. An artificial contaminant is added at a constant rate to a continuously recirculating oil system and the resultant increase in differential pressure is plotted against the weight of contaminant added, as shown in Fig. 17. The resulting curve has a characteristic form which is constant for a given filter media.

In the first stages of dirt addition, it can be seen that the pressure drop increases slowly, whereas later the pressure drop increases very rapidly. This illustrates why very little element life is gained by allowing the filter to operate at a high differential pressure after the 'knee' of the curve has been passed. The curve also illustrates the irrelevancy of a system designer concerned solely with clean pressure drops; the more significant factor is the differential pressure across the filter after a specified amount of dirt has been added.

Except with non-bypass filters, the maximum pressure drop across the element is usually determined by the bypass valve setting. The system designer must also check that the system performance is maintained with the pressure drop at its maximum value. If filters are oversized, the bypass may pass a substantial flow without indicating.

It is commonly thought that to obtain longer service life from a given filter element without sacrificing filter efficiency, it is only necessary to

Fig. 17: Typical dirt capacity curve for hydraulic filter element.

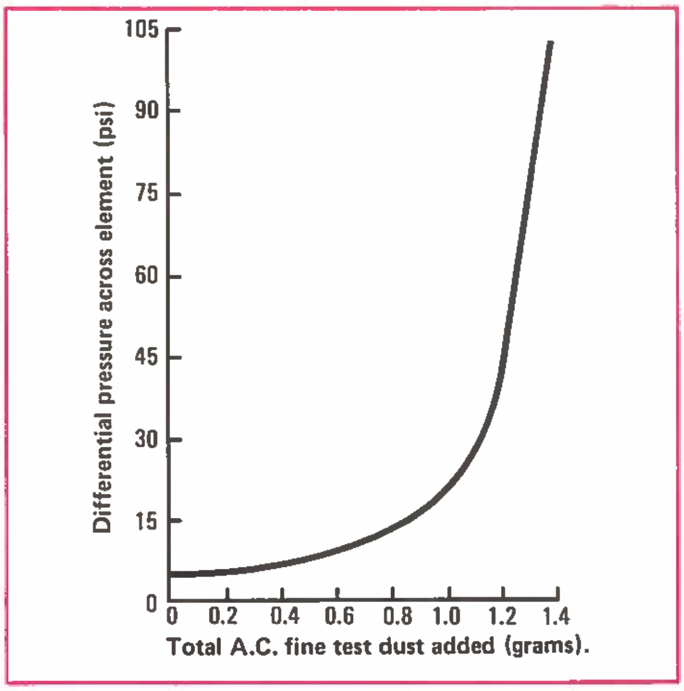
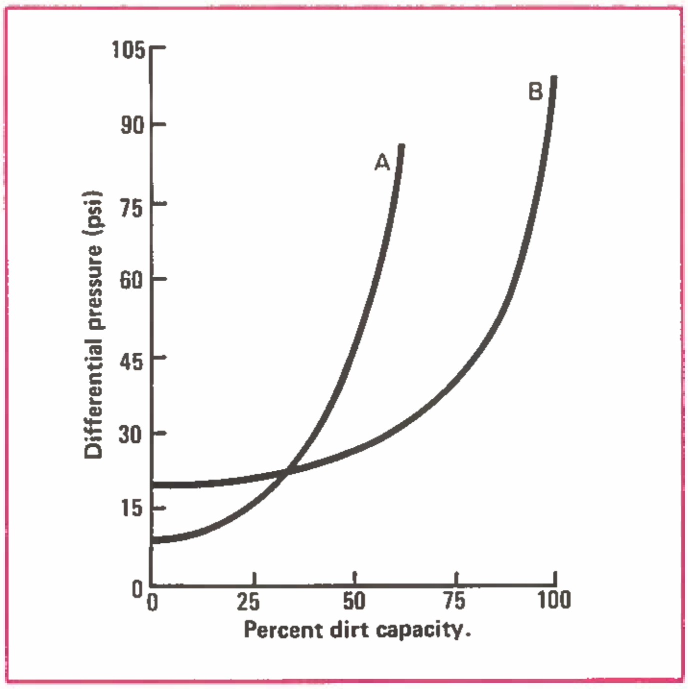


Fig. 18: Comparison of dirt capacities and differential pressures for two different filters.



pack more media area into the filter envelope. This is not so, since an optimum area exists for a given envelope, and it is detrimental to exceed this area.

The comparative life test previously described is used to compare the dirt holding capacity of different filter elements, and in Fig. 18 we compare the dirt capacity of two filters of identical envelope size. Filter A has a lower clean pressure drop than filter B because A has more area. However, the optimum area has been exceeded and, therefore, filter A would have a shorter life than B for a given pressure differential.

How close packing reduces the effective area is illustrated in Fig. 19 where the pleats close up under pressure and the small angles between them clog rapidly. Fatigue failures can also occur when pressure is applied. As data on the results of dirt capacity tests are not readily available, we must revert to the manufacturer's flow rating at a specific clean pressure drop and use this as a basis for assessing dirt capacity.

Re-capping, our objective is to achieve a 'dirt-in equals dirt-out' condition. The desired dirt capacity of the filtration system will therefore depend to a large extent on our ability to control the dirt input. The dirt input is a product of inbuilt contamination and ingressed contamination, which in turn produces system generated contamination. Let us first examine each source in detail and the factors controlling its input (Fig. 20).

Based on Fig. 20 we will now grade the cleanliness level of a system between 1 and 7. An example of a grade 1 clean system would be a clean workshop with effective control over all contamination ingress. A grade 7 dirty system would be a foundry with little or no control over contamination ingress, and a system operating several exposed cylinders.

Fig. 21 will assist in making a numerical assessment between these values, based on the environment and the degree of control over contamination.

We must now relate this to the effective element area and the maximum allowable pressure drop. The relationship between area and pressure drop is not simple, but by using very broad approximations of these values, and assuming that the manufacturer's

Fig. 19A: Increasing filter area within a given package may actually reduce the dirt holding capacity.

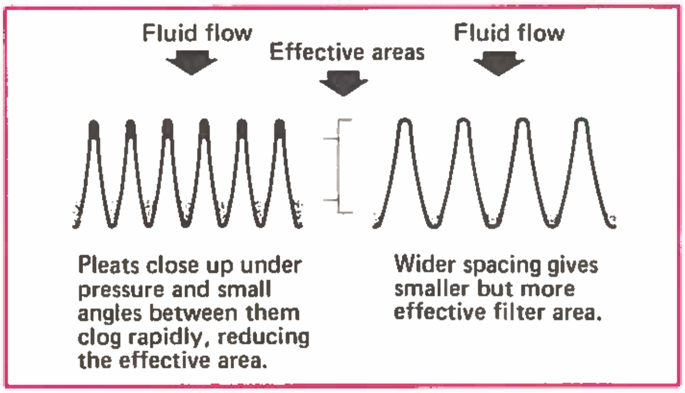
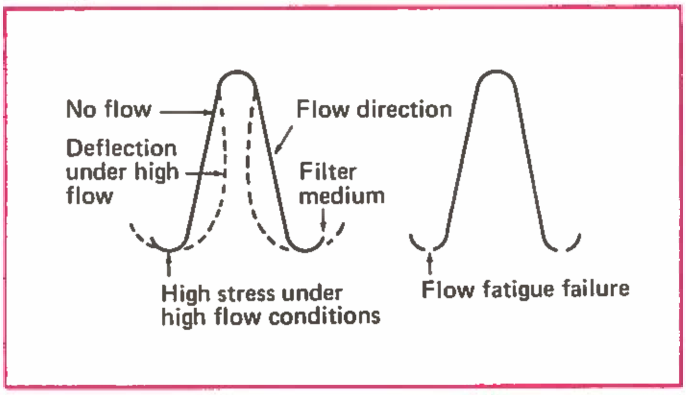

Fluid flow
Effective areas
Fluid flow
Pleats close up under pressure and small angles between them clog rapidly, reducing the effective area.
Wider spacing gives smaller but more effective filter area.

Fig. 19B: Changes in flow and pressure drop cause the sides of element corrugations to flex and the root to stretch, giving rise to fatigue stresses.


No flow
Deflection under high flow
Flow direction
Filter medium
High stress under high flow conditions
Flow fatigue failure

flow rating at a specific clean pressure drop is a good guide to dirt capacity, the following selection guides can be used: Fig. 22 for pressure line filters, Fig. 23 for return line filters, and Fig. 24 for off-line filters.

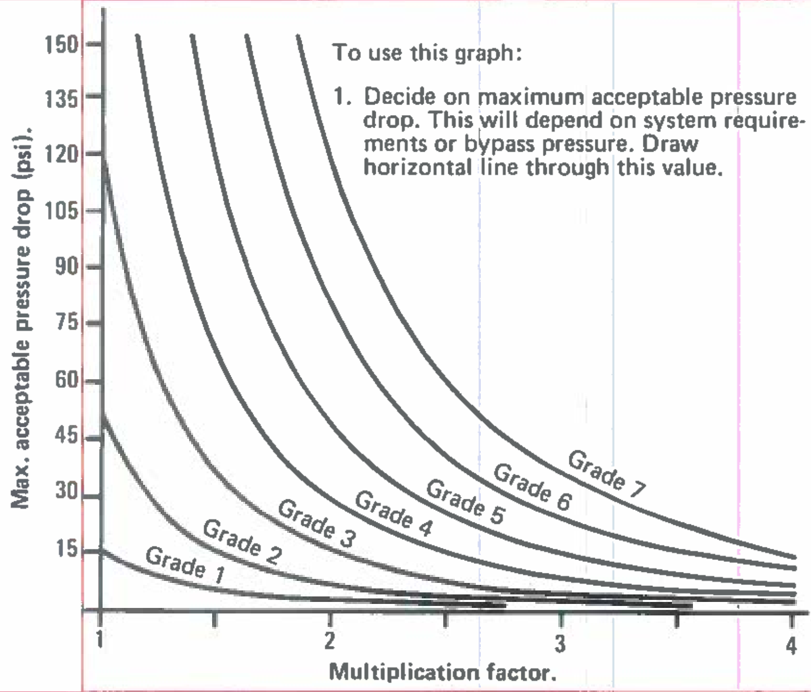
Fig. 20: The practical steps which control contamination in hydraulic systems.

Contamination source	Controller
Inbuilt in components, pipes, manifolds, etc.	Good flushing procedures, system not operated on load until acceptable contamination level obtained.
plus Present in initial charge of fluid	Integrity of supplier. Fluid stored under correct conditions (exclusion of dirt, condensation, etc.). Fluid filtered during filling.
plus Ingressed through air breather	An effective air breather with rating compatible with degree of fluid filtration.
plus Ingressed during fluid replenishment	Suitable filling points which ensure some filtration of fluid before entering reservoir.
plus Ingressed during maintenance	This task undertaken by responsible personnel. Design should minimize the effects.
plus Ingressed through cylinder rod seals	Effective wiper seals or, if airborne contamination, rods protected by suitable gaiters.
plus Further generated contamination produced as a result of the above and the severity of the duty cycle.	Correct fluid selection and properties (viscosity and additives) maintained. Good system design minimizing effects of contamination present on system components.

Fig. 21: Assessment and classification of system cleanliness level into 7 grades.

ENVIRONMENTAL CONDITIONS			
GOOD	AVERAGE	BAD	DEGREE OF CONTROL
3	6	7	Little or no control over contamination ingression (many exposed cylinders).
2	4	5	Some control over contamination ingression (few cylinders).
1	2	3	Good control over contamination ingression (gaitered cylinders).

Fig. 22: Pressure line filter selection guide.

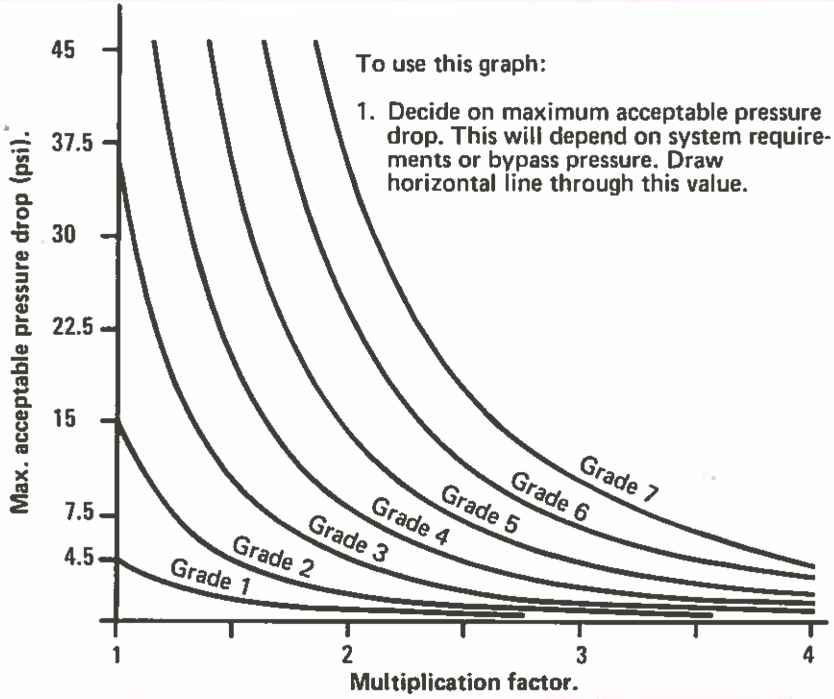


To use this graph:

1. Decide on maximum acceptable pressure drop. This will depend on system requirements or bypass pressure. Draw horizontal line through this value.
2. Assess cleanliness grade in system, from Fig. 21. Draw vertical line through intersection of pressure drop and cleanliness grade line.
3. Read off multiplication factor where vertical line crosses horizontal axis. Multiply this factor by the actual flow rate at chosen pressure line location. Now select a filter element to handle this revised flow rate at a pressure drop of 15 psi (according to manufacturer's clean rating).

Actual flow x multiplication factor
= recommended filter capacity.

Fig. 23: Return line filter selection guide.

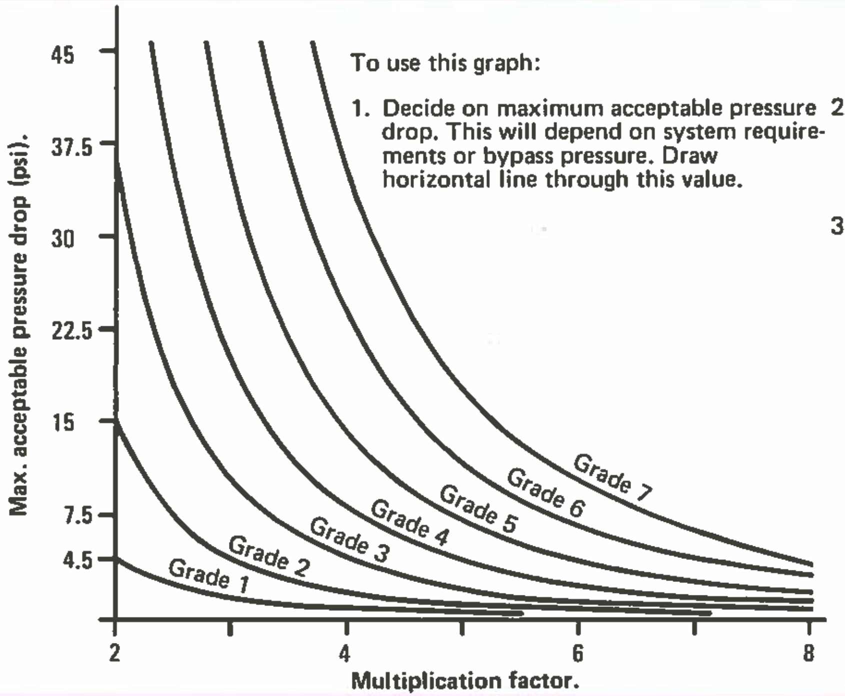


To use this graph:

1. Decide on maximum acceptable pressure drop. This will depend on system requirements or bypass pressure. Draw horizontal line through this value.
2. Assess cleanliness grade in system, from Fig. 21. Draw vertical line through intersection of pressure drop and cleanliness grade lines.
3. Read off multiplication factor where vertical line crosses horizontal axis. Multiply this factor by the actual flow rate at chosen return line location. Now select a filter element to handle this revised flow rate at a pressure drop of 4.5 psi (according to manufacturer's clean rating).

Actual flow x multiplication factor = recommended filter capacity.

Fig. 24: Off-line filter selection guide.



To use this graph:

1. Decide on maximum acceptable pressure drop. This will depend on system requirements or bypass pressure. Draw horizontal line through this value.
2. Assess cleanliness grade in system, from Fig. 21. Draw vertical line through intersection of pressure drop and cleanliness grade lines.
3. Read off multiplication factor where vertical line crosses horizontal axis. Multiply this factor by the actual offline pump flow rate. Now select a filter element to handle this revised flow rate at a pressure drop of 4.5 psi (according to manufacturer's clean rating).

Note: To achieve reasonable life a minimum multiplication factor of 2 is recommended.

Actual flow x multiplication factor = recommended filter capacity.

CHAPTER 6

Locating the Filter

Pump Inlet Filtration

Correct filling of the pump is vital if satisfactory operation of the hydraulic system is to be achieved. All too often, insufficient attention is paid to the total pump inlet configuration and, as a result, cavitation remains a major cause of pump failure.

For this reason it is worth looking once more at the basic principles of pump filling. The most common method is to use atmospheric pressure acting on the fluid surface of the reservoir to force fluid into the pump inlet chambers. For convenience, the pump is often sited above the fluid level, Fig. 25.

From basic hydraulic principles we know that for flow to take place we must have a pressure difference. With this arrangement we rely on the mechanical action of the pump to create a depression at its inlet. Manufacturers usually quote a maximum allowable

Fig. 25: Negative head tank.

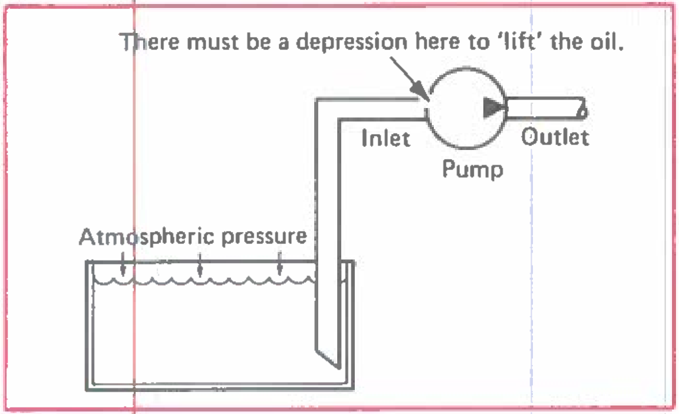
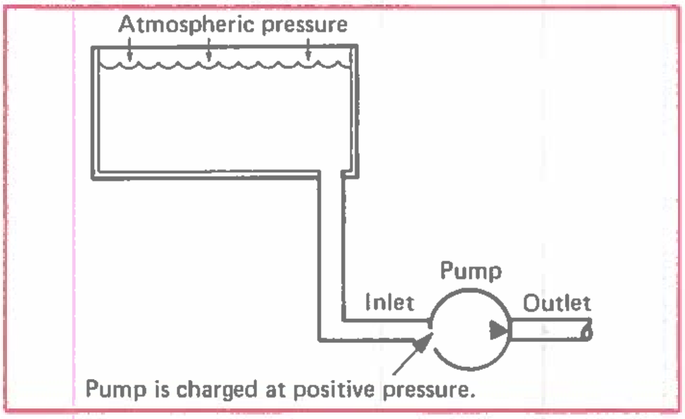


Fig. 26: Positive head tank.



depression at the pump inlet, which is likely to be of the order of 5" Hg. Thus with normal pressure drops accounted for, only a very small pressure drop can be tolerated across the filter. For this reason the size and cost of inlet filters is often greater, for example, than filters in the return line. Furthermore, such low pressure drops make silt removal virtually impossible.

Fire resistant fluids are very sensitive to suction pressures. They have higher specific gravities than mineral oils, particularly the synthetic and HWBF (95% water, 5% oil) types. This increases the pressure drop to the pump and at the same time demands a higher pressure to accelerate the fluid into the pump. Water glycols and water-in-oil emulsions have a high vapor pressure and the pump inlet depression should be limited wherever possible to half the value for mineral oils, even when temperature is limited to 120°F. With or without inlet line filters, it is usually essential to provide a positive head at the pump inlet when using these fluids, Fig. 26.

Regardless of the fluid type, a positive head will improve inlet conditions because it increases the force available to create the required flow.

For both negative and positive head inlet arrangements, when sizing inlet filters we must ensure that any filter (housing and element) and associated pipework: 1) pass the full pump volume within the permitted inlet depression for that pump, and 2) permit a bypass flow that is still within that limit when the filter element is blocked.

All calculations should take into account the effect of higher viscosity fluid, e.g. at cold starts; otherwise, cavitation will occur.

The usual micrometre rating for inlet filters is 74 or 149. However, units are available incorporating elements down to 10 micrometre. This means large housings and probable oversizing of the inlet configuration.

The 74 and 149 micrometre mesh elements will remove most of the particles above their rating but are relatively ineffective in removing anything smaller. The addition of magnets will remove some of the fine metallic particles. The location of the magnets within the filter, however, must be such that under no circumstances can accumulated contaminant break away and pass a conglomerate into the pump.

An advantage often claimed for inlet filters is ease of servicing. However, incorrect re-assembly of access covers can result in air ingress which often goes undetected and can be harmful to the system.

Inlet filters are generally used in systems where maintenance procedures do not prevent quantities of large particles entering the reservoir, e.g., when topping up. If bypassing does not occur, they protect the pump from this type of contamination.

It is an encouraging sign that more designers are providing separate filling arrangements, usually through some form of coarse filter to reduce the need for last-chance pump inlet protection.

If the filling requirements of the pump are critical and supercharging is necessary (as is the case quite often with large variable-displacement piston pumps), it is common to locate a filter between the supercharge and main pumps. For such applications, the same guidelines given for the sizing of pressure or return line filters can be used. However, depressions can occur when a variable pump moves from zero to full displacement, and the likely effect on filter performance must be carefully considered.

Pressure Line Filtration

Let us start by discussing the location of the pressure line filter in relation to pump and relief valve.

Fig. 27 shows the pressure filter located downstream of the relief valve. For the non-bypass type the arrangement shown in Fig. 28 is mandatory. The actual flow seen by the filter during the operating cycle depends on the system demand, and during off-load periods there is virtually no flow if block-center directional valves are employed. Naturally, if off-loading is achieved through open-center directional valves, the filter will see full pump output for this period.

To maintain constant flow across the filter, it has become common practice to locate the pressure filter between pump and relief valve, Fig. 29. Advocates of this arrangement point out that the relief valve is then protected from pump-generated dirt. For this arrangement a bypass is mandatory, and the filter manufacturer must assure that any filter malfunction will not result in excessive pressure at the pump outlet.

Where variable-displacement pumps are employed, careful analysis of the flow handled by the filter must be made. Take the classic case of a pressure compensated pump which is compensating (i.e. low displacement at maximum system operating pressure), where

Fig. 27: Pressure line filtration with bypass filter.

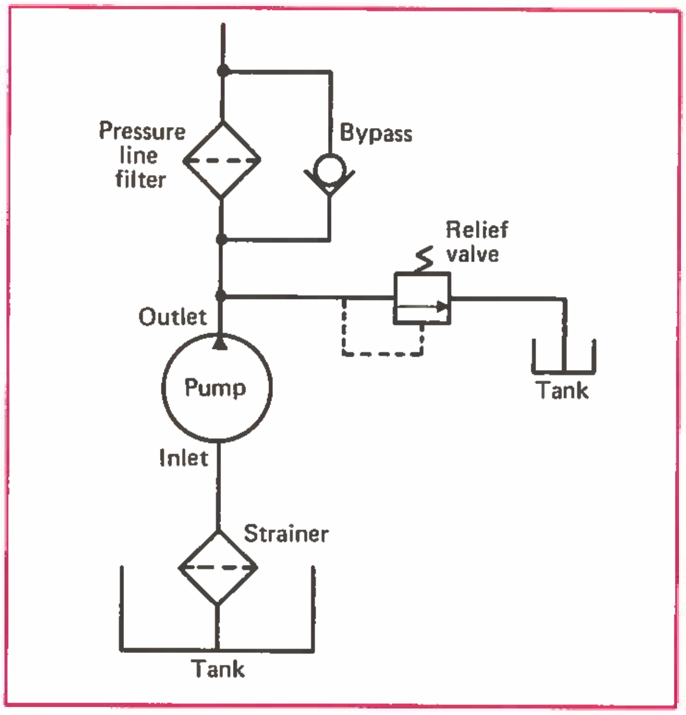
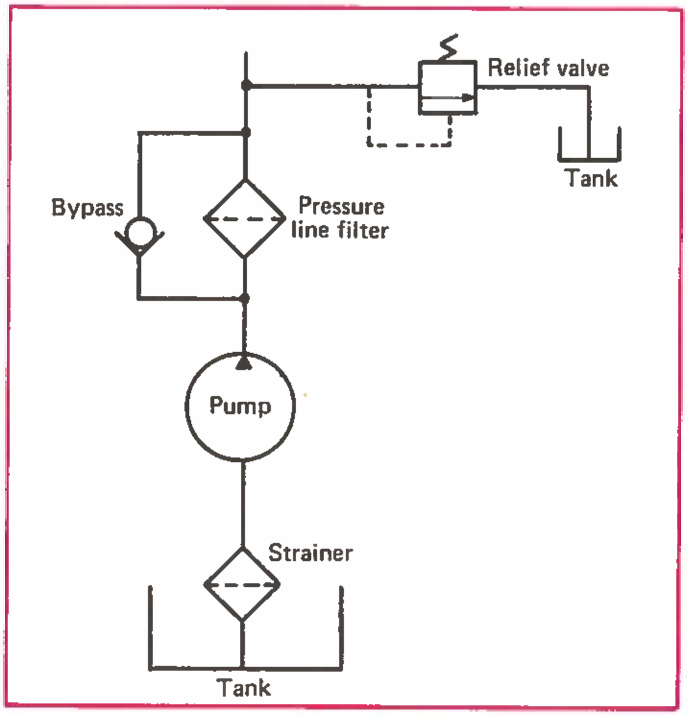


Fig. 29: Locating pressure filter before relief valve gives constant flow through filter.



a low flow rate exists although the pump is still operating at pressure. With a pressure line filter, dirt removal is limited. Even when flow demands are made, these are limited to cylinder displacement.

As illustrated by Fig. 15 in Chapter 5, filter efficiency depends upon the type of flow it sees, although currently no standard test exists for evaluating filters when they are subjected to abrupt flow and pressure changes and mechanical vibration. Practice has shown there is a reduction in efficiency, and its extent depends not only on the quality of the actual filter media but also on how well this media is supported,

Fig. 28: Pressure line filtration with non-bypass filter.

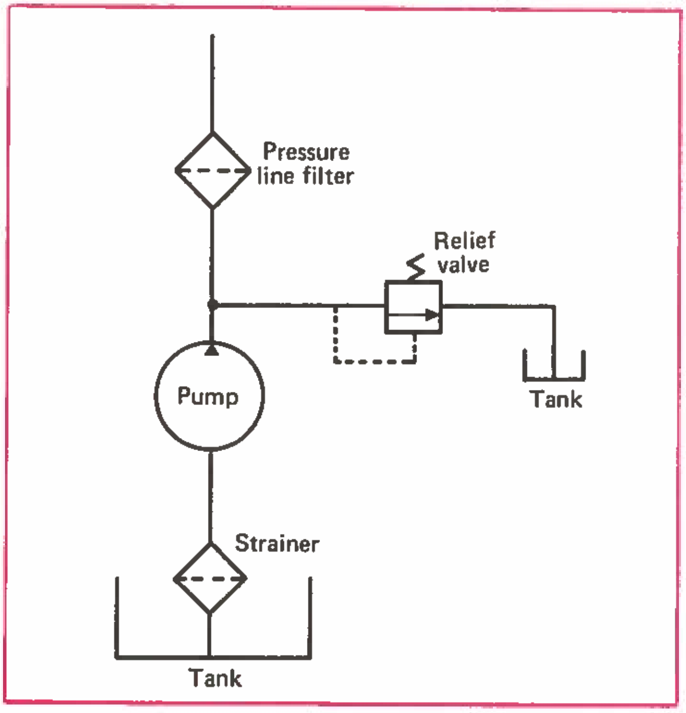
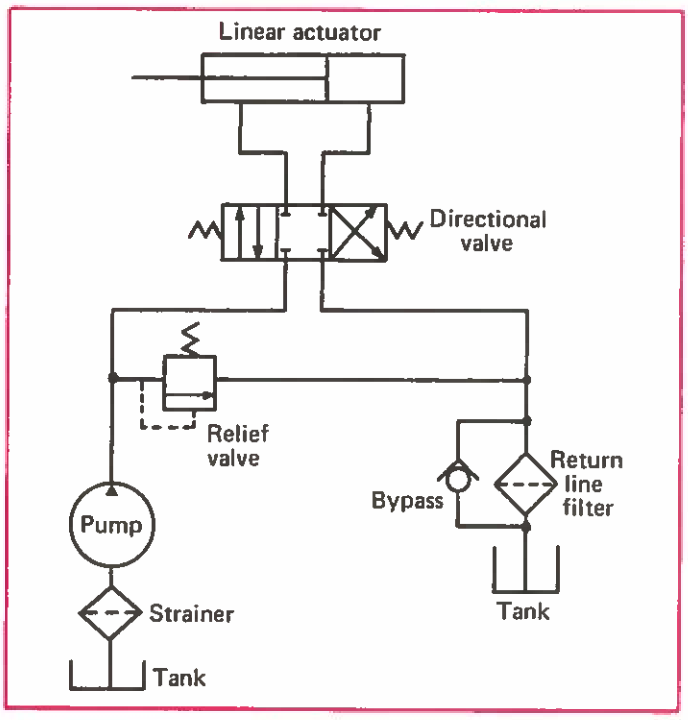


Fig. 30: Basic arrangement of return line filtration.



and bypass valve operation. In the future it will be essential that the system designer has information on the performance of integral bypass valves under system operating conditions.

In addition to mechanical vibration and abrupt flow and pressure changes when valves are operated, a pressure line filter is also subject to pump pulsations. These effects were demonstrated on a hydraulic component test rig where the so-called 10 micrometre nominal filter proved almost totally ineffective in removing sub-5 micrometre contaminant.

Because pressure filters have to withstand the full system pressure with an adequate safety margin, there is a tendency to make these small, thus limiting their dirt holding capacity. It is usual for manufacturers to quote a rated flow at around 15 psi for bypass types, and at a slightly higher pressure for non-bypass types which incorporate high pressure differential capability elements.

In summary, it can be said that pressure filters are subjected to conditions far removed from those found in a laboratory filter test rig and, because of this, the designer's task in assessing the resultant contamination is extremely difficult. The levels given in Fig. 15, Chapter 5, are based on field experience using good quality elements.

Pressure filters may well be used in certain applications to protect the system should catastrophic failures of the pump occur, or to provide special protection to a single unit or group of units. An example of the latter would be a servo valve, where failure might be extremely expensive, though it should be noted that a pressure filter does not protect a servo valve from dirt ingressed through cylinder rod seals. In such cases, a non-bypass filter should be considered with the assurance that total element collapse cannot occur. If bypass types are employed, an indicator should be provided to warn of a partially blocked element.

From a maintenance point of view, changing pressure filter elements involves stopping the system unless external bypass valving is provided.

Return Line Filtration

The usual return line filtration arrangement shown in Fig. 30 has all return lines passing through the filter. Drain lines from pumps, motors and certain valves should not be subjected to pressure surges emanating from the system return lines, and should return separately to tank.

Where there are high surges (e.g. due to uncontrolled decompression or the rapid acceleration of the fluid column in the relief valve tank line when this valve operates) it may be undesirable to pass these through the filter. To prevent collapse of the element due to high-viscosity oil, e.g. at a cold start, or when the element is loaded with dirt, an internal or external bypass should be provided.

When bypassing occurs under minimum flow and surge conditions, the circulated fluid should not be contaminated by dirt already retained in the filter. The type of surge experienced in most return lines will reduce the filter efficiency and, in the absence of precise data, the filter should be selected using the pulsating flow condition from Fig. 15, Chapter 5.

Full flow return filtration should be of sufficient capacity to handle the maximum return flow (including any in excess of pump flow where, for example, unbalanced cylinders are used) without the bypass opening.

Careful assessment of the flow across the filter is necessary. The comments relating to the use of variable displacement pumps with a pressure filter again apply since, with pressure compensated pumps, the filter sees only the volume displaced by the cylinder.

For certain systems, partial return line filtration is acceptable with the bypass (internal or external) always passing a percentage of the flow. With this arrangement much depends on the continuous rate of flow which we can get across the filter. Satisfactory performance can sometimes be achieved with as little as 10% passing across the actual element.

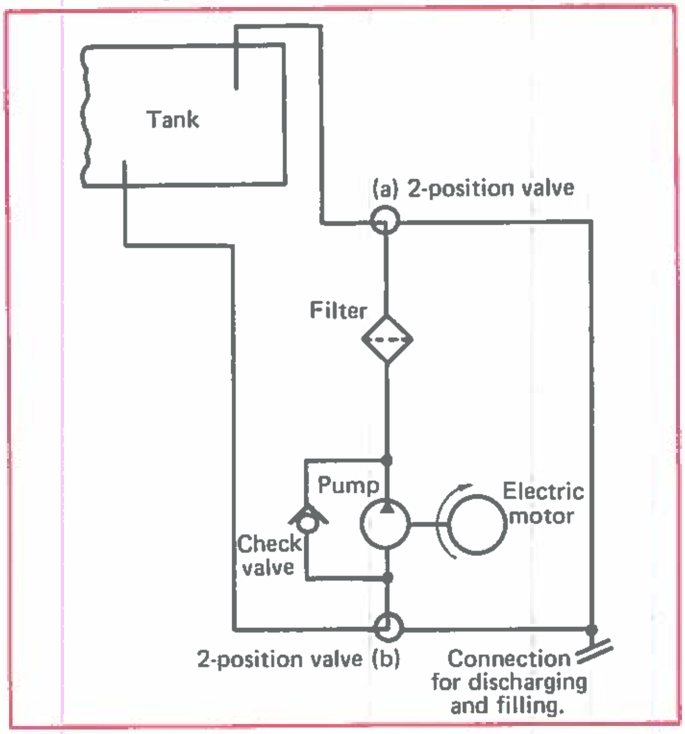
To summarize, a return line filter does not protect the system from environmental dirt entering the reservoir via breathers or when topping up. However, if we start off with a clean reservoir and take precautions to prevent the ingress of environmental contamination, then experience has shown that effective filtration can be achieved economically with return line filters.

Off-Line Filtration

It has been stated that the effectiveness of filters located in both pressure and return lines is reduced by shocks, surges, pulsations, vibrations, etc. to an extent which depends on the type of media and how well it is supported. Steady flow relatively free of pressure fluctuations provides optimum filter performance. The simplest way of achieving this is to remove the filter from the main system and place it in an independently powered circulating unit where its performance is more predictable. Though connected to the system reservoir, a typical arrangement is shown in Fig. 31. The likely contamination level from our three filter ratings under steady flow conditions is given in Fig. 15, Chapter 5, and our ability to provide the correct flow rate will determine whether or not these levels can be achieved and maintained.

Assuming reasonable standards of engineering in the design and construction of the system, the most

Fig. 31: Layout of off-line filtration system. Valves (a) and (b) can be positioned so that the unit will provide a filtered fill or discharge.



relevant factors in selecting the flow rate through off-line filtration are likely to be environment and tank size. From field experience to date, the guidelines given in Fig. 32 can be applied and the flow rate derived can be used to determine the size of the filter by the procedure outlined in Chapter 5.

With off-line filtration, the designer is not governed by the flow and pressure characteristics of the main system. The best filter, flow, and size can be readily selected to achieve the desired frequency of maintenance.

Should the desired contamination level not be achieved, then corrections are easily made to the flow rate or type of filter without affecting the design of the main system. Furthermore, the off-line installation can be run prior to starting the main system in order to clean the oil in the tank and reduce the contamination level the pump will see at start-up. By the addition of simple valve gear, it can be employed to filter the initial charge of fluid and any used subsequently in topping up. Ideally it should be left running continuously to provide a complete tank of clean fluid ready for every start-up.

Unlike line filters, the off-line installation will continue to clean up the fluid when variable delivery pumps are running at minimum displacement.

Being independent of the main system, off-line filters can be placed where they are most convenient for servicing. When element changes are necessary the main system is not affected; the operation can be carried out at any time without stopping or introducing air into the main system, thus making the very minimum of servicing skills acceptable.

Whether or not off-line filtration can be the sole means of filtration depends on many factors related to the character, quantity and origin of the contamination. As it is partial filtration, we must decide if it is necessary to protect individual or groups of components from stray particles likely to cause a catastrophic failure.

Fig. 32: Suggested flow rates for off-line filtration.

Environment	Flow rate (l/min) as percentage of tank capacity (litres).
GOOD	5%
AVERAGE	10%
BAD	20%

Summary of Filter Location

The foregoing has attempted to show that there is a place for inlet, pressure, return and off-line filtration.

Surely what is most important is to realize that, like all branches of engineering, system filtration engineering involves taking a risk. A single particle in the 5 to 10 micrometre size range, for example, could cause system malfunction if it arrived at the wrong place at a critical time. The machine tool designer may well have the benefit of a prototype on which to evaluate the filtration system performance and make changes prior to the first production batch. The designer of heavy steelworks equipment must be right the first time and may have to use all the filter selection procedures available in order to minimize the risk of expensive stoppages later. That risk will be with us until a) we have more realistic data on filter performance, and b) maintenance practices are greatly improved.

The preceding chapters can best be summarized diagramatically (Fig. 33). This highlights that control over the inbuilt contamination by applying good installation practices, and using all means available to limit the amount of ingressed contamination, enables us to maintain the system-generated contamination at an acceptable level.

The level of contamination entering the pump is a critical factor, and should be controlled to prevent the sort of action depicted in Fig. 33. The difficulty in controlling this level by inlet filters has been clearly stated. They should be used only to prevent large particles from entering the pump and causing catastrophic failure.

Downstream of the pump, the ability of a pressure filter to trap particles will be influenced, to a large extent, by any abrupt changes of flow and pressure which have the effect of driving particles through the filtering media. We know from practice that bypass valves can malfunction under certain circumstances, and it is essential that the design of the filter prevents any migration of contamination.

Ingress through cylinder rod seals, although in the smaller particle size range, can nevertheless add up to significant quantities. The performance of any return line filter will, again, depend upon the magnitude of flow and pressure changes.

Off-line filtration provides optimum filter performance, thus enabling us to control contamination levels more effectively. Our ability to remove contamination depends not only on the filter micrometre rating, but also on the flow rate across it, and we must provide sufficient dirt holding capacity to ensure an element life acceptable to the end user.

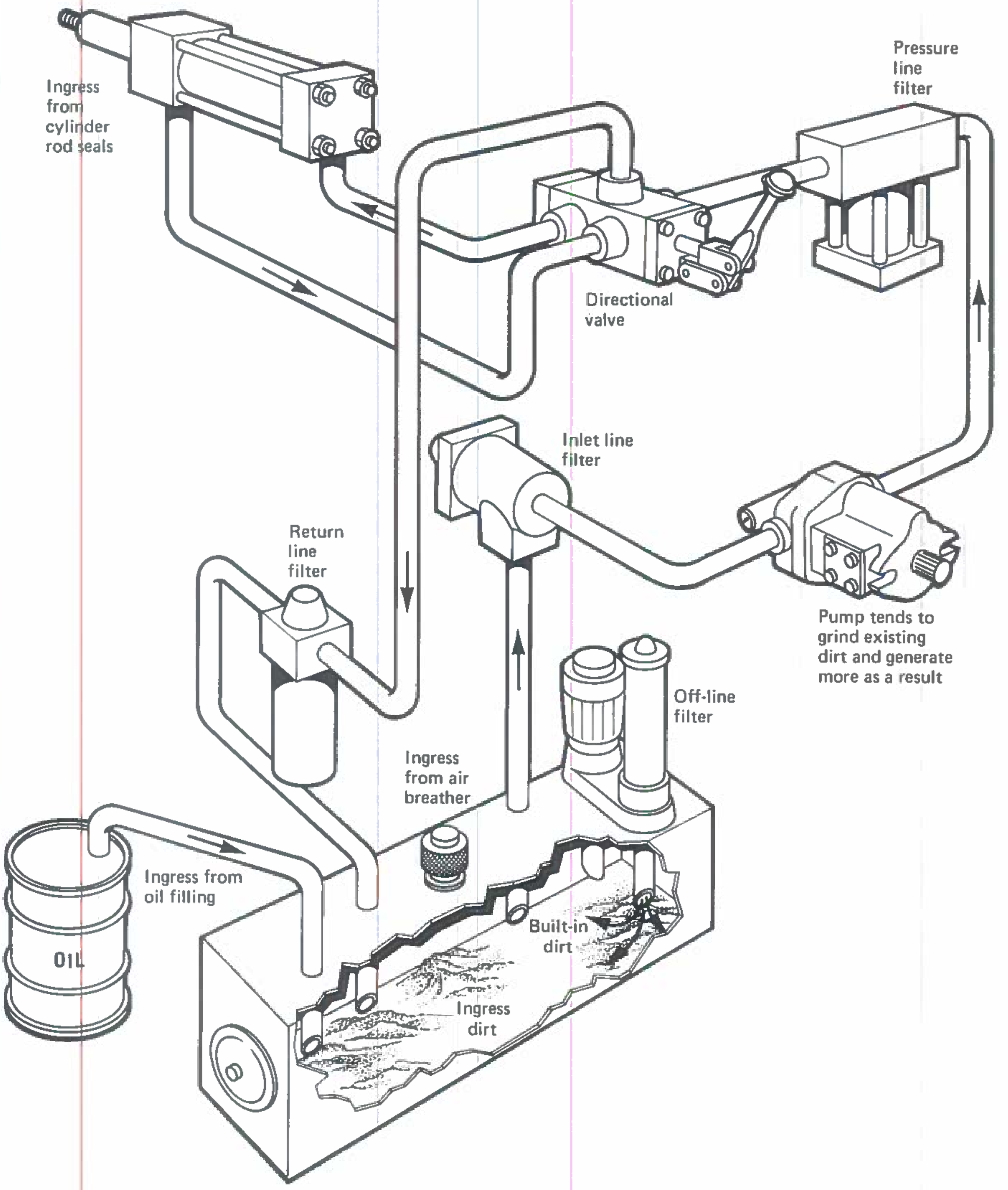
The key design steps have been extracted from the text, and are presented in Chapter 7.

CHAPTER 7
Design Steps and Worked Examples

Design Steps

1. Identify the critical component or components. As discussed in Chapter 3, the most critical component in many systems is the pump. However, spool valves which are infrequently operated and subjected to continuous pressure need protection from high silt concentrations. Flow controls at low settings, irrespective of pressure, must be protected against silt. Flow and pressure control valves give greater repeatability when the silt content is under control. Using the cleanliness level chart, Fig. 10, we must define the dirt sensitivity of the system components, i.e.
   a) Very sensitive components, e.g. proportional control units.
   b) Average hydraulic components, including most pumps.

Fig. 33: A pictorial representation of the basic filtration equation.



Total system dirt = Built-in dirt + Ingress from oil + Ingress through breather + Generated in cylinder + Ingress from cylinder rod seal + Generated in pump + Generated in valve − Dirt removed by line filters − Dirt removed by off-line filter

c) Very tolerant components, e.g. those operating at low pressures.

Knowing the system pressure, the desired contamination level (band width) can be determined.

2. From Fig. 15 determine which category of filter under both steady and pulsating flow conditions is required. This will be just one of the factors to consider when locating the filter or filters in terms of quantity, type and size.
3. Make a detailed assessment of all potential contamination ingress points (see Fig. 20). Fig. 20 can help in determining what to look for. By protecting the system more effectively from ingressed contamination we may increase the initial costs, but in the longer term reduce running costs; it is total costs which are most important. Each means of controlling contamination listed in Fig. 20 must be carefully examined and then the system may be graded by using Fig. 21.
4. Select filter location or locations, again keeping in mind that the best economic solution must be based on both initial and running costs.
5. Determine flow rate at each filter location. Remember variable displacement pumps such as the pressure compensated type may spend considerable periods of time at low displacement, but under full working pressure; therefore, the pump still needs protection. Flows in excess of maximum pump displacement may pass through the filter, e.g. exhaust flow from the full bore end of a differential cylinder. Uncontrolled fluid decompression can result in very high flows for short periods.
6. Determine maximum acceptable pressure drop at each filter location. This should take into account all operating conditions such as cold starts, maximum flow rate, fluid viscosity, etc.
7. Analyze the type of flow at the chosen location; in the pressure or return line it is likely to be pulsating (Fig. 15). Off-line location can be considered as steady flow. The actual filter rating can now be decided.
8. Sizing filter.
   a) Pressure filter, use Fig. 22.
   b) Return line filter, use Fig. 23.
   c) Off-line filter, use Fig. 24.
9. Select suitable filter condition indicator to meet user's stated requirements. (Find out these requirements. Do not assume them.)
10. On a continuously operating machine, provide properly designed facilities for changing the element without stopping the machine. (Not necessary for off-line filtration).
11. Ensure filters are readily accessible for maintenance.
12. Make sure that the inclusion of in-line filters will not adversely affect the operation of components.
13. Ensure that the filter media is compatible with the system fluid.
14. Consider long-term operation. In large systems particularly, provide means for flushing sections where fluid is likely to be trapped.
15. Provide sampling points.
16. Prepare recommendations for contamination control by the user. Methods of recording data and analyzing results are discussed in Chapter 8.

Worked Examples

The following examples show how the various graphs and charts are used once the filter location has been established.

Example 1. Pressure Line Filtration

Maximum system pressure, 2000 psi.
Fixed displacement pump output, 8 gpm.

Dirt sensitivity of components. Small sliding spool valves subjected to pressure and therefore possible silting.

From cleanliness level graph (Fig. 10) take the line for average components at 2000 psi. Recommended profile 17/13.

From Fig. 15 select filter to achieve 17/13 level at 5 and 15 μm, bearing in mind possible silting. Choose partial silt remover.

The environment is considered good but with little or no control over contamination ingress, since system uses cylinders. Recommended environment rating is Grade 3 (Fig. 21).

Pressure across element limited by bypass to 45 psi. From pressure line filter selection guide (Fig. 22) intersection of Grade 3 and 45 psi gives multiplication factor of 1.4.

Summary. System requires partial silt removal filter which will pass 13.7 gpm at a clean pressure drop of 15 psi at the predicted system fluid viscosity.

Example 2. Return Line Filtration

Maximum system pressure, 1000 psi.
Fixed displacement pump output, 26 gpm.

Dirt sensitivity of components. The pump, valve and cylinder would be classed as having average dirt tolerance.

From cleanliness level graph (Fig. 10) take the line for average components at 1000 psi. Recommended contamination profile is 19/15.

Now, from Fig. 15, choose a filter to achieve this level. In this instance a chip remover could be used but, since a relief valve discharges through the filter and there are also likely to be decompression surges, a partial silt remover is recommended.

The maximum flow through the filter will be 52 gpm with a 2:1 area ratio cylinder.

The environment is considered good with some control over dirt ingression. Suggest Grade 2 is used (Fig. 21).

Pressure across element is limited by filter bypass to 22 psi. From return line filter selection guide (Fig. 23) it is clear that the intersection of Grade 2 and 22 psi would give a multiplication factor of less than 1. In such cases the minimum factor of 1 would be used. In this case we are providing sufficient capacity to take the flow when the full-bore end of the cylinder is discharging; during an off-load period the filter will see a flow of only 26 gpm.

Summary. For this application we need a filter which can be classified as a partial silt remover and which will pass 52 gpm at a clean pressure drop of 4.5 psi at the predicted system fluid viscosity.

Example 3. Off-Line Filtration

Maximum system pressure, 3000 psi.
Variable displacement, pressure compensated pump.
Maximum output, 18 gpm.
Reservoir capacity, 60 gallons.

Dirt sensitivity of components. The critical component would be the pump.

From cleanliness level graph (Fig. 10) take the line for average components at 3000 psi. Recommended contamination profile is 15/11.

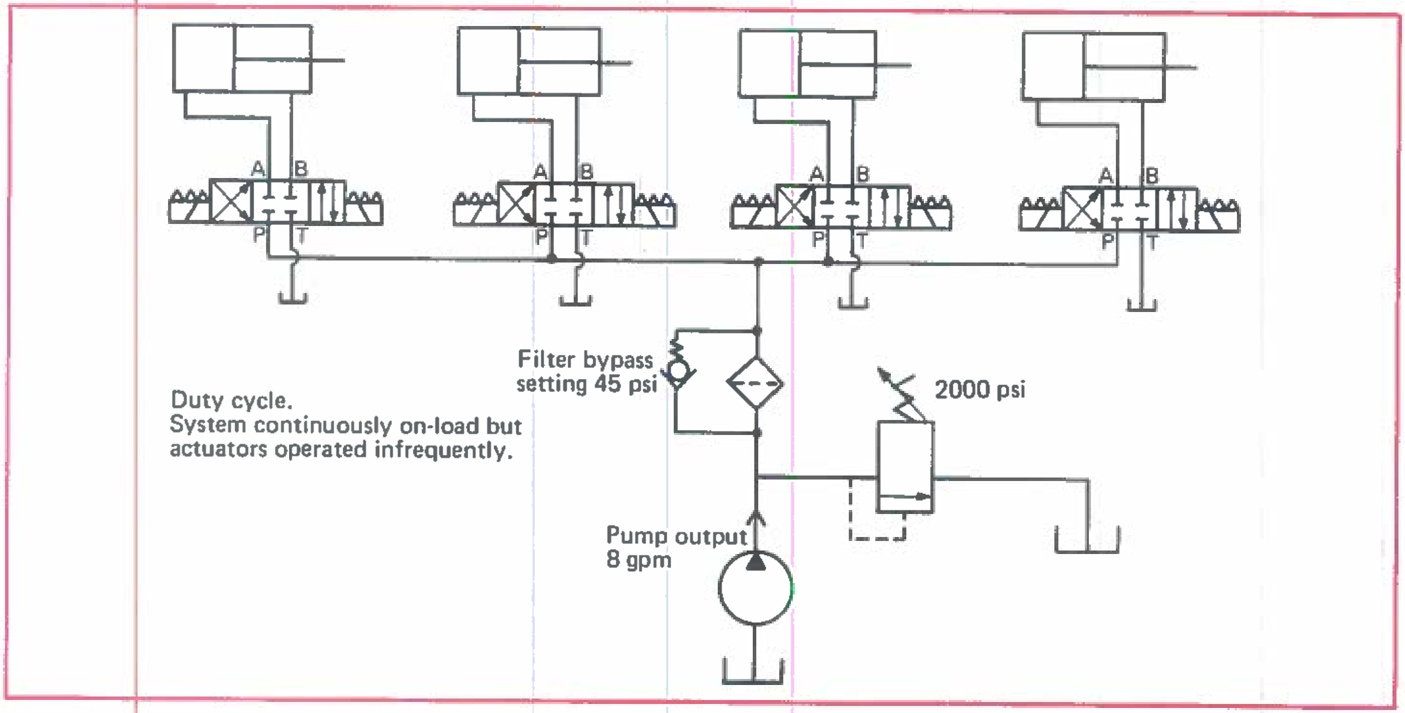
From Fig. 15 choose a filter to achieve this level. We can see that, under the steady flow conditions that apply in this case, the 15/11 level is likely to be achieved by either a silt or partial silt remover. It then becomes a question of risk. In practice it is a simple operation to change from one category to another on site, so the risk is minimal.

From Fig. 32, assuming average conditions, off-line pump flow rate is 6 gpm.

The environment is considered bad but, by using recommended design practices, a good control over ingression is achieved. Assume Grade 3.

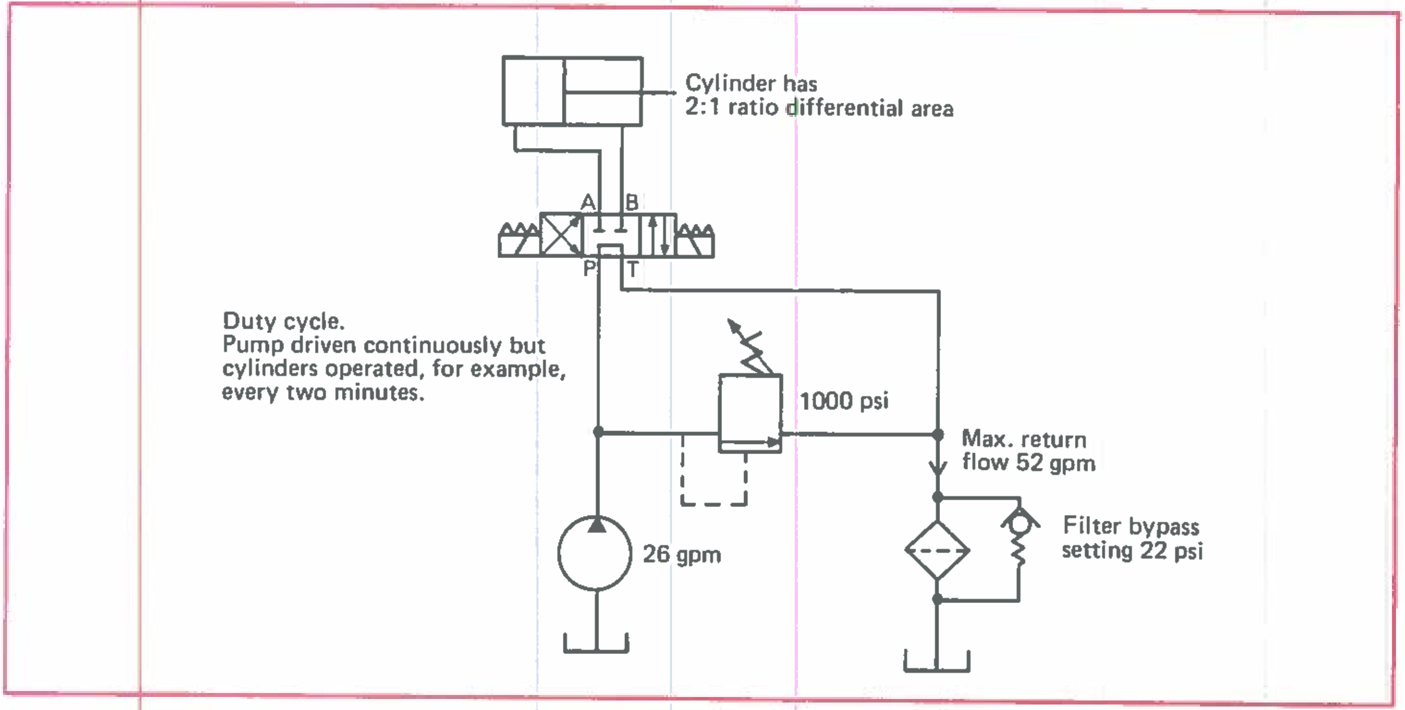
Pressure drop across filter element is limited by filter bypass to 15 psi. From off-line filter selection

EXAMPLE 1: Pressure line filtration



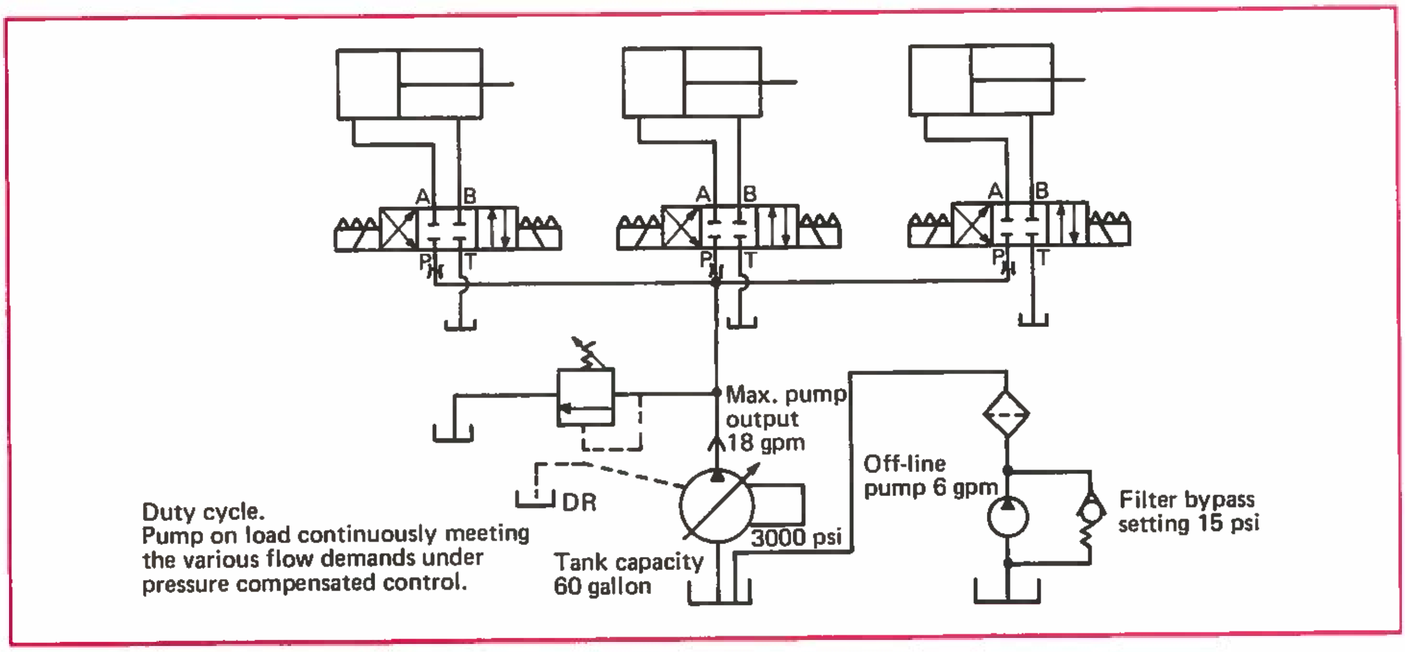
Duty cycle.
System continuously on-load but
actuators operated infrequently.

EXAMPLE 2: Return line filtration



Duty cycle.
Pump driven continuously but
cylinders operated, for example,
every two minutes.

EXAMPLE 3: Off-line filtration system



Duty cycle.
Pump on load continuously meeting the various flow demands under pressure compensated control.

guide, Fig. 24, intersection of Grade 3 and 15 psi gives a multiplication factor of 2.7.

Summary. For this application the off-line system would consist of a 6 gpm pump and a partial silt remover filter, which could be changed easily to a silt remover if the desired cleanliness level was not attained. The element should be able to pass 14 gpm at a clean pressure drop of 4.5 psi and at the predicted system fluid viscosity.

CHAPTER 8
Fluid Sampling and Analysis

Very few fluid power systems have valves for taking fluid samples. However, in most cases, reservoir samples can be taken at a point where the fluid enters the pump inlet. Where sampling valves are provided within the system, they can be used to monitor the actual filter performance.

Wherever the sampling valves are located, they must provide a truly representative sample. At present, there are no standard procedures for sampling a hydraulic system; until there are, a syphoning procedure for tank sampling can be used. If a permanent sampling valve for tank sampling is to be fitted, it should be a ball valve with a Teflon* seat and be piped in the manner shown in Fig. 34. This type of valve adds a minimum of contamination to the sample taken.

If other sampling points are necessary they should preferably be where turbulent flow conditions exist, and a tee type installation without a probe is often used. However, there appears to be a difference of opinion among experts on this particular aspect. The same type of ball valve should be used, and it should be installed in a line at some point in the system where normally there is a constant flow rate.

Since only about 200 ml of fluid is to be collected, it is important that the valve be as small as possible. The ideal is to use a small valve and collect the sample with the valve fully open, and with a fine bore tube downstream to restrict the flow rate. If a large valve is used, it will have to be almost completely closed to make the flow rate low enough for practical filling of the sampling bottle. The small fluid passages through the valve in this condition will usually be small enough to remove larger contaminants, and erosion can occur. Both effects will destroy the validity of the sample.

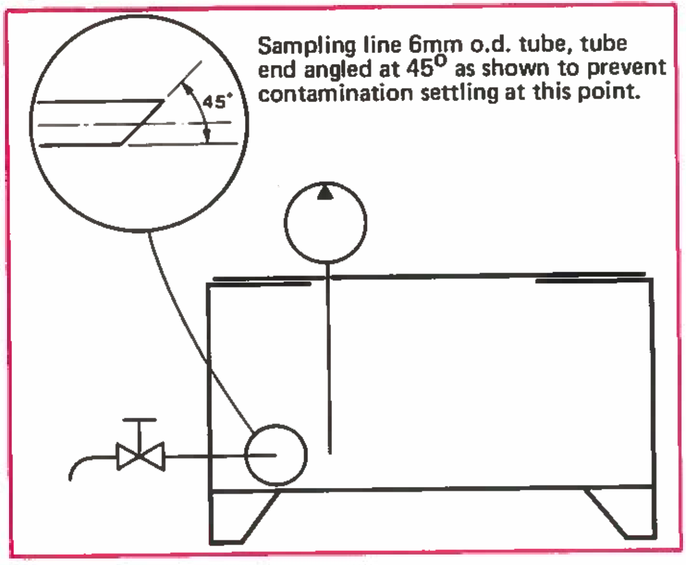
Considerable care has to be exercised in this operation to obtain reliable results. The following procedures have evolved from experience.

Bottle Cleaning

'Nominally clean' new bottles with screw-on caps are used. Then:

1. Add approximately 25 ml of clean solvent and shake well.

Fig. 34: Position and detail of tank sampling point.



Sampling line 6mm o.d. tube, tube end angled at 45° as shown to prevent contamination settling at this point.

Registered trademark of E.I. DuPont DeNemours Co., Inc.

2. Empty bottle immediately after shaking.
3. Repeat 1 and 2.
4. Rinse inverted bottle thoroughly using a suitable clean solvent.
5. Repeat 4.
6. Place clean plastic film over mouth of bottle and screw on cap.

Three Sampling Procedures

There are three established methods for taking fluid samples, and the following imperatives apply to all of them.

Never obtain a sample by breaking a line, e.g. by cracking a joint or draining a hose.

Always take the sample under the cleanest conditions possible.

Method 1 is used if there is a suitable valve for sampling, remembering the earlier remark concerning flow.

1. Operate the system for at least 30 minutes before taking the sample, to distribute the contaminant as evenly as possible within the fluid.
2. Open the sampling valve and flush at least one pint of oil through the valve.
3. Place the sampling bottle under the valve while it is still open and collect 150 to 200 ml of fluid. Do not touch the valve while taking the sample – it could release particles from the valve that are not representative of the contaminant in the system.
4. Withdraw the sampling bottle and shut the valve.
5. Cap the sampling bottle immediately.

Method 2 is used if a suitable valve is not available and a standard valve of unknown contamination shedding characteristics must be used.

1. Operate the system for at least 30 minutes prior to taking the sample.
2. Open the valve and flush at least 2 gallons of oil through the valve before the sample is taken. This is best accomplished by connecting the outlet of the valve to the oil reservoir by using plastic tubing. Do not close the valve.
3. After flushing the valve remove the plastic tubing from the valve, while the latter is still open and flowing oil, and fill the sample bottle with 150 to 200 ml of fluid. Do not touch the valve while taking sample.
4. Withdraw the bottle, then close the valve.
5. Cap the sampling bottle immediately. If at all possible, valves with graphite-filled stem packing should be avoided because they often continue to leak graphite into the oil. This method is unsuitable for pressures above 200 psi.

Method 3 is used in hydraulic systems where no sampling valve is fitted and it becomes necessary to take a reservoir sample. Provided the following preparations and procedure are carried out, a representative 'particle count' from the sample will be obtained.

The equipment comprises a suitable length of plastic tubing with a 2-foot length of glass tube at one end and a 3-inch length at the other, pre-cleaned 250 ml sampling bottles, and an aerosol solvent which will dissolve oil.

Procedure

1. Before sampling, the tube should be sprayed internally and externally with the solvent.
2. The sample should be taken when the system is at its normal working condition, or at least 30 minutes after start-up.
3. The reservoir breather or cover should be removed, ensuring first that the surrounding area is perfectly clean. The positioning of the sample tube in the reservoir is important – it should be at least 24 inches from all fluid boundaries. The sample is then drawn by suction into the sampling tube and transferred to the sample bottle. The operation is repeated until the required quantity of fluid has been collected.

Analyzing Particulate Contamination

Particle count methods are suitable for particles larger than 5 micrometre and they also give some indication of the type of contaminant. Either microscopes or automatic particle counters can be used.

Of the microscope methods, the ARP-598A, or a modified version of it, is widely employed. The fluid sample is filtered through a laboratory membrane and the particles retained on the membrane surface are sized and counted statistically.

The counting is very time consuming, and there can be variations when different people count the same slide. However, provided expert counters are used, the variations are generally within acceptable limits. A trial count should be carried out to compare the readings of the operators.

Automatic Counters

There are at least two modern machines employing scanning computers and TV screens. The contamination is presented on a pre-prepared membrane and a greatly magnified image is displayed on the face of a monitor. The field of particles is scanned and all individual particles, or particles with selected characteristics, can be picked out.

They are counted and the results are displayed as numbers. The computer side of the instrument can be programmed to give this information in any form, such as number per given volume of oil or percentage distribution. The digital information stored can be converted into cross section area or the longest dimensions on the x or y axis.

Other instruments use the principle of light interception to count particles suspended in a liquid. Bottle samples can be used, or the instrument may be connected direct to a hydraulic system, since the sampling head can withstand line pressures to at least 3000 psi. The sample fluid flows through a small rectangular passage and past a window. Particles pass the window one at a time, provided specified limits of concentration are not exceeded. Light from a tungsten lamp is formed by the window into a parallel beam of exact cross section and directed onto a photodetector. As it passes the window, each particle interrupts an area of the light beam according to its size. It does not matter whether the light has been absorbed, reflected or refracted as long as it does not reach the photodetector.

This causes a specific reduction (or pause) in the voltage that is proportional to the size of particle. Several counting channels with preset thresholds sum the particles by size, and a size range adjustment is provided for each channel to permit the operator to select any desired range.

As these instruments operate on the blockage of light principle, they measure the projected area of a particle and then record the diameter of a circle of equivalent area. Unfortunately, they cannot differentiate between solid particles and air bubbles.

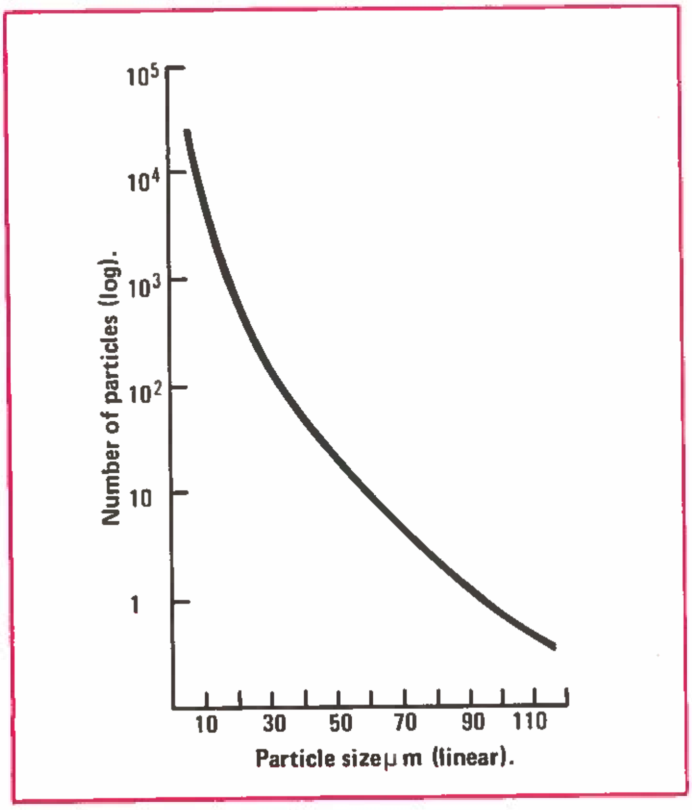
At the present state of the art, it is not possible to accurately correlate the results from the various counting methods in use. Therefore, it is imperative when comparing methods that experts in these various methods and techniques be employed.

Particles Smaller than 5 Micrometre

There are problems when counting particles smaller than 5 micrometre by the light interception method. Smaller particles can be detected by the microscope, but they are very difficult to count.

There is a method termed Silt Index which gives a non-quantitative measurement of particles smaller than 5 micrometre. It compares the rate of flow of a clean oil through a membrane with that of the sample fluid at the same pressure. The limitation of this method is that any unmixable liquid such as water has the same effect as particles, i.e. it clogs the membrane, thereby increasing the Silt Index Number.

Fig. 35: Typical particle count plotted on a log/linear graph.



Reporting of Particle Contamination Data

For graphic representation many of the existing classification standards use a log/linear relationship (see Fig. 35) with the number of particles plotted on a cumulative basis. This type of graph tends to accentuate the larger sizes of particles, which represent a very small percentage of the total, and tends to crowd together particles below 15 micrometre. In order to examine the relationship of the smaller particles in greater detail, the graph can be replotted on an enlarged scale using a log/log^2 plot (Fig. 36).

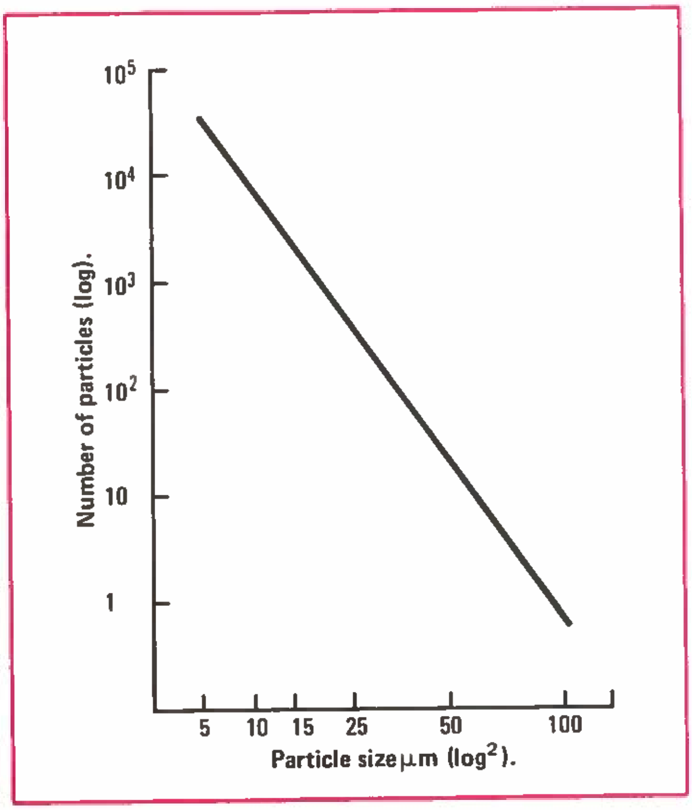
The task of producing contamination classes from such data would be made much easier if straight lines with common slopes were assumed. As we have seen previously, this is unlikely to occur in practice. (For example, the slope will depend not only on the element but also on its location in the system.)

What is needed then is a more logical and simple system of numbering which takes into account the limits of measurement accuracy and more accurately portrays the different slopes. It also has to be a simple, effective and consistent means of communication between suppliers and users of hydraulic equipment, whereby the level of solid contaminant in the fluid can be clearly described.

The ISO S.C. CODE system goes a long way to fulfilling these requirements and a summary of its relevant sections is given here.

The profile is intended for use in fluid specifications, but the standard is not intended to recommend or specify the acceptable levels in any specific system. It does not specify or recommend a method of measurement, and neither does it specify the type of contaminant.

Fig. 36: The curve of Fig. 35 re-plotted on log/log^2 axes to give a straight line relationship.



Most existing methods of defining solid contaminant quantities are based on the supposition that all such contaminants have a similar distribution of particle size. This may be true of natural contaminants such as airborne dust. It is not necessarily true for contaminants which have been circulated in a system, subjected to crushing in pumps and to separation in filters.

To allow for such changes in distribution, the profile is defined by two numbers indicating, respectively, the number of solid particles above 5 micrometre and 15 micrometre per 100 ml sample of fluid.

In order to keep the number of ranges to a reasonable minimum and still ensure that each step is meaningful, a step ratio of two has been used and Fig. 37 shows how each quantity has been allocated to a range number.

The procedure used is as follows:

Using a 100 ml sample, first count all the particles above 5 micrometre and allocate a range number from the right hand column.

Next, sum all particles above 15 micrometre and again allocate a range number as with the 5 micrometre count.

For example, Fig. 38 shows the results of a typical millipore particle count.

In this case the number of particles above 5 micrometres is 200,668 and hence would have a range number of 18; similarly, the number of particles above 15 micrometres is 5,468 and hence would have a range number of 13. By combining the two numbers with a solid line we get an 18/13 profile (Fig. 39).

There are two ways in which this system can be used. The first is by adhering rigidly to the rules just stated, and the second is by comparing the results of actual measurements plotted on a graphical background.

Fig. 40 gives 34 examples of contaminant profiles which can be created easily from the table in Fig. 37.

The ISO S.C. CODE is based on the fact that hard abrasive particles sized around 5 micrometres have an apparent aspect (length to width) ratio approaching unity, and that sizes above 15 micrometres do not generally exceed an aspect ratio of 3.

Thus, additional information is needed to describe the fibrous content, the abrasiveness of the contaminant or any special contaminants. In some cases it may be essential to specify the method of measurement, e.g. microscope or type of automatic counter.

The advantages claimed for the ISO S.C. CODE system can be summarized as follows:

The prime number of the class, i.e. the quantity of particles above 15 micrometre, is produced relatively accurately by all current systems of counting, both manual and automatic.

It classifies the contaminant levels in the two most

Fig. 37: Allocation of range of numbers in the ISO S.C. CODE system to particle counts.

No. of particles per 100 millilitres		Range Number	No. of particles per 100 millilitres		Range Number
More than	& up to		More than	& up to	
8M	16M	24	2k	4k	12
4M	8M	23	1k	2k	11
2M	4M	22	500	1k	10
1M	2M	21	250	500	9
500k	1M	20	130	250	8
250k	500k	19	64	130	7
130k	250k	18	32	64	6
64k	130k	17	16	32	5
32k	64k	16	8	16	4
16k	32k	15	4	8	3
8k	16k	14	2	4	2
4k	8k	13	1	2	1

M = million; k = thousand

Fig. 38: Results of a typical particle count.

Particle size range	No. of particles per 100 ml of oil
5 – 15 micrometres	195,200
15 – 25 micrometres	3,880
25 – 50 micrometres	1,280
50 – 100 micrometres	232
Above 100 micrometres	76

Fig. 39: The contaminant code is constructed by allocating a range number to the total number of particles above 5 micrometres, allocating a second range number to the number of particles above 15 micrometres, and then combining these range numbers with a diagonal thus: 18/13.

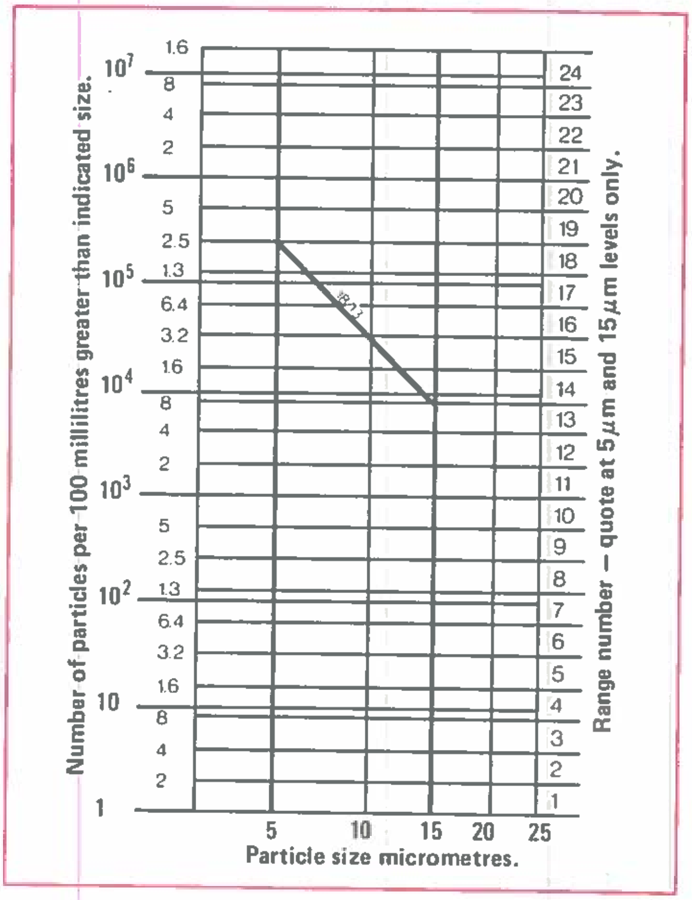


Fig. 40: Tabular presentation of contamination levels and corresponding codes.

CODE	Number of particles per 100 millilitres			
	Over 5 µm		Over 15 µm	
	More than	& up to	More than	& up to
20/17	500k	1M	64k	130k
20/16	500k	1M	32k	64k
20/15	500k	1M	16k	32k
20/14	500k	1M	8k	16k
19/16	250k	500k	32k	64k
19/15	250k	500k	16k	32k
19/14	250k	500k	8k	16k
19/13	250k	500k	4k	8k
18/15	130k	250k	16k	32k
18/14	130k	250k	8k	16k
18/13	130k	250k	4k	8k
18/12	130k	250k	2k	4k
17/14	64k	130k	8k	16k
17/13	64k	130k	4k	8k
17/12	64k	130k	2k	4k
17/11	64k	130k	1k	2k
16/13	32k	64k	4k	8k
16/12	32k	64k	2k	4k
16/11	32k	64k	1k	2k
16/10	32k	64k	500	1k
15/12	16k	32k	2k	4k
15/11	16k	32k	1k	2k
15/10	16k	32k	500	1k
15/9	16k	32k	250	500
14/11	8k	16k	1k	2k
14/10	8k	16k	500	1k
14/9	8k	16k	250	500
14/8	8k	16k	130	250
13/10	4k	8k	500	1k
13/9	4k	8k	250	500
13/8	4k	8k	130	250
12/9	2k	4k	250	500
12/8	2k	4k	130	250
11/8	1k	2k	130	250

The above tables cover the most usual series of codes between ranges 8 and 20. Other codes which are not shown above can be constructed from Fig. 39.

important zones describing the relationship of coarse and fine particle quantities.

It allows for and indicates differing slopes between the 5 and 15 micrometre counts.

A classification can be made direct from counts without the need to compare with a graph, although the latter will show benefits in certain cases.

It does not greatly conflict with any existing system.

The ratios chosen allow the most common contaminant ranges to be described with two figures, yet the ranges are adequately spaced to have useful significance.

Fig. 41 shows a commonly used format for displaying particle count data.

CHAPTER 9
Conclusion

Any examination of the subject of contamination involves four groups of people:

1. The fluid manufacturer or supplier.
2. The hydraulic equipment and filter manufacturers.
3. The manufacturer of the machinery which uses the hydraulic equipment.
4. The end user of the machinery.

Each of these has a commercial responsibility to supply equipment which will perform its duty satisfactorily at a reasonable price, and each must have some knowledge of the cleanliness of the working fluid.

The fluid supplier will supply fluid as clean as required and will charge accordingly.

The hydraulic equipment manufacturer should recommend the types of fluids, and their degree of cleanliness, best suited to the hydraulic equipment recommended for a specific application.

The machinery manufacturer must be aware of all the conditions to be met, including availability of fluids, maintenance practices and the machine reliability required. It is the manufacturer's job to offer customers the best value. This could mean, for example, providing either an inexpensive throwaway filter which must be replaced relatively frequently, or a more expensive one for which the only service needed is the occasional exchange of a relatively inexpensive component, such as a filter cartridge.

The end user has to make the final judgement, by comparing the real value of each machine offered; each user will place different emphasis on the values of the equipment offered, depending on the job to be done. To the end user, the value of a machine is its fitness for the purpose, how long it will satisfactorily perform, and what the cost of servicing will be. He is not interested in how many 10 micrometre particles are contained in 100 ml of hydraulic fluid.

The user's interest is in the least-expensive filter that will provide the required degree of cleanliness. In making this assessment, the original cost of the equipment has to be balanced against the cost of service. For example, the advantages offered by a very expensive pump that will operate on 'dirty' oil may have to be compared with those of a low-cost pump plus filter.

If it is to maintain its present high integrity, the hydraulic equipment industry needs to establish more meaningful specifications for filtration. The specifications must allow the end user to buy performance without necessarily having to know how this performance is achieved. The responsibility for establishing specifications must be divided between manufacturers of filter elements, who know what is possible, and the manufacturers of hydraulic equipment, who know what is needed. These two groups should be able to communicate in meaningful terms.

At present there are no adequate techniques covering all aspects of contamination measurement in hydraulic fluids. This should not deter us from making a start in the right direction. The cleanliness level chart, for example, will not be 100% right the first time; there will be a continual need to review it in the light of new experience.

The escallating costs of equipment downtime and maintenance are encouraging the end user to consider operating costs more carefully, for both new and existing equipment — along with the part that contamination control plays in determining those costs. It is hoped that this volume has given a useful insight to the real nature of the problem and has pointed the way to a systematic approach and more cost-effective solutions.

Fig. 41: Suggested format for displaying particle count data.

PARTICLE COUNT RESULTS SHEET

CUSTOMER:

MACHINE:

SAMPLE:

SAMPLE DATE:

SAMPLING METHOD:

FLUID:

COUNTING METHOD:

RESULTS:

PARTICLE SIZE, MICROMETRES	> 5	> 15	> 25	> 50	> 100
PARTICLES PER 100 ml					

ISO SOLID CONTAMINATION CODE: /

CONTAMINANTS OBSERVED AS ESTIMATED PERCENTAGE OF TOTAL CONTAMINATION:

BRIGHT METAL %
BLACK METAL %
SILICA %
ELASTOMERS/PLASTICS %
COTTON FIBRES %
CARBON %
OTHER (SPECIFIED) %

OTHER OBSERVATIONS: (E.G. WATER CONTENT P.P.M.)

ANALYZED BY:

DATE:

VICKERS

A TRINOVA COMPANY

VICKERS, INCORPORATED
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For the location of our nearest distributor or district office, dial toll-free 1-800-243-8160.

348A-K87D

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VICKERS FILTER FACT SHEET

Typical Model Code	Nominal Micronic Rating	Rated Flow (GPM)	Maximum Pressure (PSI)	Beta 10 Ratio	Dirt Holding Capacity ACFTD (Grams)	Press.drop Drop at Rated Flow Clean Elem. (PSI)	Standard By-pass Pressure (PSI)	Replacement Element Kit Number
OFP-15-S-10	10	15	3000	5	2.25	15	40	926412
OFP-15-S-3M-10	3	15	3000	75	1.5	29	40	926410
OFP-35-S-10	10	35	3000	5	5.6	25	40	926416
OFP-35-S-3M-10	3	35	3000	75	2.3	35	40	926414
OFR-15-S-10	10	15	600	1.5	9.5	3	25	941447
OFR-15-S-3M-10	3	15	600	20	5.5	8	25	941445
OFR-30-S-10	10	30	600	1.5	20	8	25	941448
OFR-30-S-3M-10	3	30	600	20	9	13	25	941446
OFR-60-S-10	10	60	400	1.5	47	3	24	941411
OFR-60-S-3M-10	3	60	400	20	33	6	24	941409
OFR-120-S-10	10	120	400	1.5	100	7.5	24	941412
OFR-120-S-3M-10	3	120	400	20	65	13	24	941410
OFRS-15-S-10	10	15	100	2	17	7	25	573082
OFRS-15-S-25M-10	25	15	100	DNA	24 (coarse dust)	4	25	573083
OFRS-25-S-10	10	25	100	2	17	13	25	573082

VICKERS FILTER FACT SHEET

Typical Model Code	Nominal Micronic Rating	Rated Flow (GPM)	Maximum Pressure (PSI)	Beta 10 Ratio	Dirt Holding Capacity ACFTD (Grams)	Press.drop Drop at Rated Flow Clean Elem. (PSI)	Standard By-pass Presssure (PSI)	Replacement Element Kit Number
OFRS-25-S-25M-10	25	25	100	DNA	24 (coarse dust)	7.5	25	573083
OFRS-60-S-10	10	60	100	3	24	11	25	941107
OFRS-60-S-10-S14	10	60	100	3	49	8	25	926388
OFRS-60-S-3M-10	3	60	100	20	17	18	25	941190
OFRS-60-S-25M-10	25	60	100	DNA	40 (coarse dust)	8	25	941191

HINTS ON MAINTENANCE OF HYDRAULIC FLUID IN THE SYSTEM

VICKERS®
A TRINOVA COMPANY

Hydraulic Fluid Changes

Good maintenance procedures make it mandatory to keep the hydraulic fluid clean. A daily/weekly or monthly log should be kept on the hydraulic fluid condition.

No hard and fast rule can be established for changing the fluid because of the great variety of operating conditions. However, we do know that when filter elements are replaced frequently, service life of a system increases. Periodic testing of the fluid by the supplier is recommended to confirm suitability for continued use and to establish the correct fluid and filter element replacement interval.

Some of the considerations affecting hydraulic fluid life are: operating temperature, type of service, contamination levels, filtration, and the chemical composition of the fluid.

Fluid Recommendations

The basic recommendations for fluid are stipulated in Vickers bulletins I-286-S and M-2950-S. The fluids recommended by these bulletins give the assurance of adequate wear protection and excellent chemical stability under the most adverse operating conditions.

On mobile applications, the viscosity grade of the fluid should be changed in spring and autumn as is done with automobile engines. Hydrostatic transmissions and control mechanisms may require a different viscosity fluid. Fluid requirements are normally outlined in the original equipment manufacturers operation and maintenance manuals.

Draining The System

The system should be started and fluid heated before draining. This will lower the time it takes to drain the system and allow impurities suspended in the fluid to be removed. It is desirable to remove all fluid from the system. Bleeding of the fluid at the lowest point in the system will help in most cases.

Systems which have accumulated deposits that were not removed during draining must be flushed with a light viscosity fluid. The fluid should contain a rust inhibitor to protect metal surfaces against rust formation after draining.

When hydraulic fluid is added to replenish the system, it should be pumped through a 25 micron filter. If such a filter is not available, a funnel with a fine wire screen (200 mesh or finer) can be used. It is important that fluid be clean and free of all substances which will cause improper operation.

FLUID CONTAMINATION/AERATION—CAUSES AND EFFECTS

CONTAMINATION

A contaminated system can be the result of several factors: system design inadequate, poor maintenance of the system, poor housekeeping of the system and adverse operating conditions

A. System Design Inadequate

1. Reservoirs which cannot be cleaned.

2. Breathers that permit abrasives, inherent in the atmosphere to enter the system.

3. Poor cylinder packing design (no wiper to clean dirt from the piston rod).

4. Improper piston rod design (piston rods with poor wear characteristics).

5. Improper valving (anti-cavitation checks omitted from cylinder circuits with rapid drop characteristics).

6. Failure to provide adequate filteration.

B. Poor Maintenance of the System

1. Improper and unclean practices when adding fluid to the system.

2. Failure to clean breathers.

3. Failure to change pitted cylinder rods and worn cylinder packings.

4. Failure to use good cleanliness practices when changing system components.

5. Failure to change filter cartridges and/or fluid at proper intervals.

6. Failure to purge debris from the system after a pump failure.

C. Poor Housekeeping of the System

Surgical cleanliness is not required, however, ordinary clean practices during assembly will pay off in increased service life of the equipment.

Excessive and improper use of pipe thread sealer on lines and gaskets in the system can cause pump failures. This is especially true when a type of sealer is used that hardens.

Another source of contamination is fittings, hoses and lines which are received from a vendor uncapped. The use of brazed or welded fittings, and unpickled steel plating can also contribute to the contamination.

Vickers data sheet 1221-S covering the preparation of pipes, tubes and fittings should be referred to and followed. This will lower the possibility of premature failure due to contamination of the system.

D. Adverse Operating Conditions

From experience, we have found that machines used in a very dusty atmosphere and in windy areas require special components. For example, heavy duty breathers, chrome plated piston rods, plus frequent changes of the filter cartridges are also required.

Effects of Contamination

Contamination affects all types of hydraulic equipment adversely. Precision high tolerance parts are very susceptable to the effects of contamination. Dirty fluid causes wear which accelerates leakage and the development of heat in a system. Heat lowers the lubricity of a hydraulic fluid and causes additional wear.

If a hydraulic pump or motor should fail, the system becomes contaminated. Remove the unit for repair. The reservoir must be drained, flushed and cleaned. All hoses, lines, cylinders and valves should be inspected for wear and particles of the unit that failed. Flush all components of the complete system to remove metallic particles.

Replace filter elements. Dispose of the fluid removed from the system and fill the reservoir with clean hydraulic fluid. Install a new or rebuilt unit and start-up the system. Allow the system to run for a period of time to verify normal operation. Filter elements should be changed after 40 or 50 hours of operation. This guarantees that the system is essentially clean and free of any residue of the failed unit.

A very good reference catalog on contamination is available from Vickers called ''Effective Contamination Control in Fluid Power Systems''. This catalog describes types and sources of contamination, effects of types and sizes of particles, specifying contamination levels, selecting a filter, locating a filter, design steps and worked examples and fluid sampling and analysis.

AERATION (air bubbles in the fluid)

Causes

The following are candidates for the formation of air in a system.

A. Leaking inlet lines.

B. Control valve ''O'' rings leaking.

C. Shaft seal leakage.

D. Leaking cylinder packings caused by cavitating cylinders.

E. Turbulence or sloshing in the reservoir.

F. Vortexing fluid in the reservoir.

G. Release of air suspended within the fluid.

Effects

Aeration can be in many forms: large bubbles, foam or in various degrees of suspension. It usually causes pump noise (cavitation). Small bubbles cause extreme and rapid ring wear, with corresponding vane tip wear. Larger bubbles cause vanes to collapse and pound. This pounding effect develops rippling in the ring and the ring will have a dull appearance. This is more apparent on straight vane rings which are hardened cast iron. With extreme aeration cases, the wear is so rapid that a ring and vanes can be destroyed within an hour. In many cases, a large step will be worn in the ring contour at the pressure quadrant. When the step reaches a depth where the vane extends and locks, the vane and/or ring will break. Also, the shaft can break where it enters the rotor if the torque is great enough.

Cures

A. Leaking Inlet Lines

1. Pipe threaded fittings can be porous. Use an approved type of pipe thread sealer on all pipe threads.

2. If the pump inlet flange surface is rough, scored or mutilated, air leakage past the "O" ring seal can result.

With any of the above defects, air can be pulled into the system.

B. Control Valve "O" Rings Leaking

"O" rings are used to seal against port leakage in many control valves. These seals can be checked by applying heavy grease around the part to be checked. If the noise stops, the trouble has been located and repair can be initiated.

On systems which have been operating at excessive high temperatures, the "O" rings can harden and take a set. If this occurs, air leakage can result. This is true not only in a pump, but also in the rest of the components of the system. Another factor enhancing air leakage is the actual fluid composition. Fluids which have a high sulphur content tend to accelerate "O" ring hardness. This is one of the principle reasons for keeping system operating temperatures down. Normal operating temperature of a system is 90° above ambient. When operating temperatures are in excess of this value, trouble may result. Maximum operating temperatures should be checked at the pump outlet port.

C. Shaft Seal Leakage

Most vane pumps are internally drained. The shaft seal cavity is connected to the pump inlet. Excessively high inlet vacuums can cause air leakage at the shaft seal. The maximum vacuum measured at the pump inlet should not exceed five inches of mercury.

Shaft misalignment can increase the probability of air leakage past the shaft seal. Universal jointed couplings or splined couplings can cause seal leakage if not properly aligned. Straight (direct) couplings should never be used.

The use of the wrong type of tools can cause distortion or mutilation of a shaft seal at installation. The outer diameter of the shaft should be lightly polished before installation to remove any burrs or roughness in the area of the shaft seal.

Shaft seals must be made of the correct material for a given application. A material that is not compatible with system fluid can deteriorate and result in a leakage problem.

D. Leaking Cylinder Packings Caused By Cavitating Cylinders

On applications where a rapid raise and lower cycle is experienced, air can enter the system through a cylinder rod seal. Vacuums in excess of 20 inches of mercury have been recorded in systems without anti-cavitation check valves. This is enough to force dirt particles past the shaft seal into the system with the air. An anti-cavitation check will allow flow from the reservoir to enter the rod area of the cylinder during a vacuum condition. Anti-cavitation checks should always be used to prevent a high vacuum condition from developing. This will lower the possibility of fluid contamination through the rod seal of a working cylinder.

E. Turbulence or Sloshing in the Reservoir

Return lines, if improperly located, can cause turbulence and aeration. A plexiglass window should be placed in the prototype reservoir to study flow conditions. Return lines emptying above the fluid level cause bubbles to form in the system. Return lines should always be terminated below the fluid level. Vehicle movement can cause sloshing within the reservoir. Reservoir must be deep enough to prevent aeration.

F. Vortexing Fluid in the Reservoir

If the fluid level in the reservoir is low and the inlet demand is great, a vortex condition can develop which pulls air into the pump inlet. In a hydraulic system, vortexing is normally the result of low fluid or poor reservoir design.

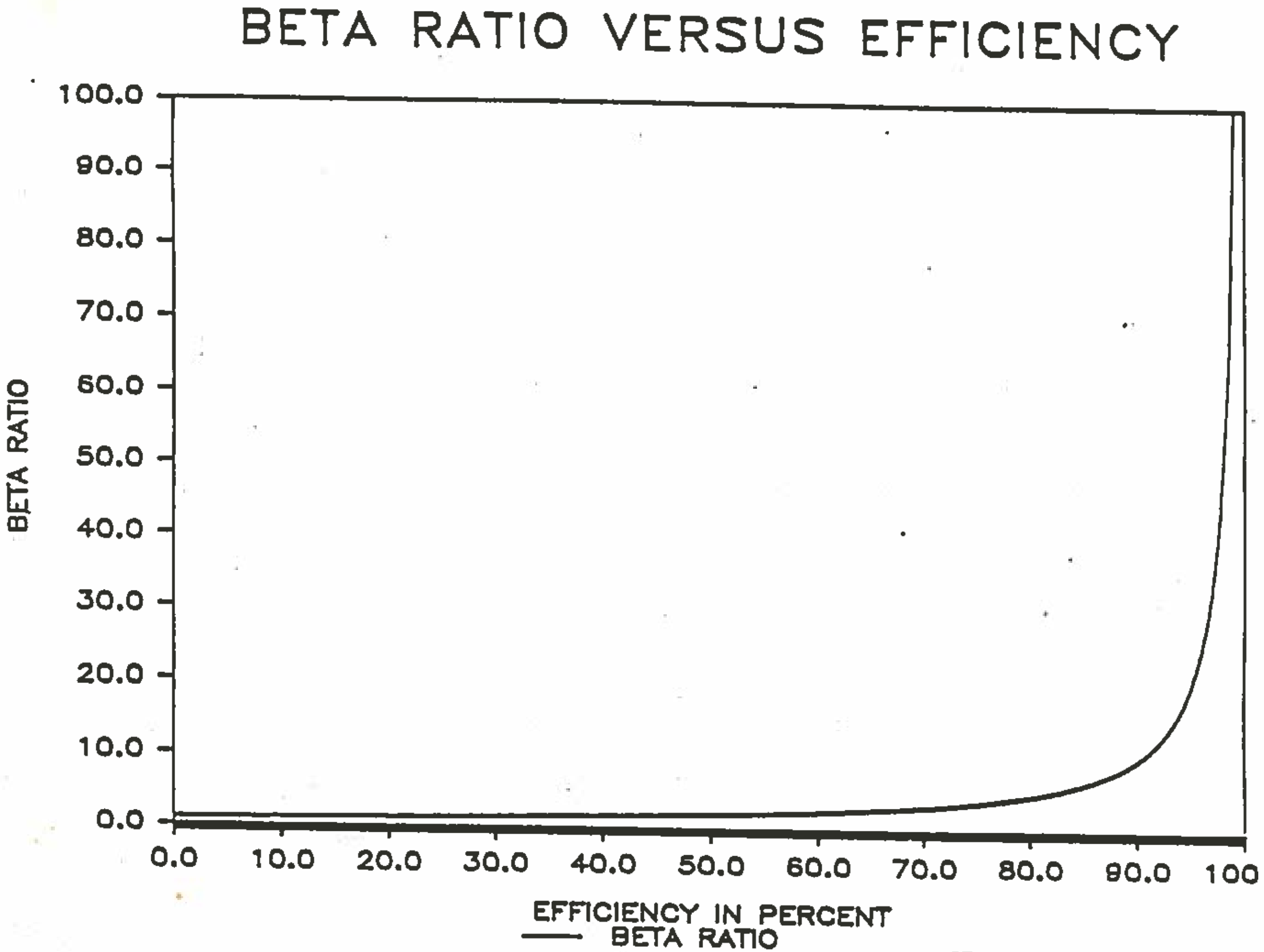
One of the best ways of curing a vortex problem is to place an anti-cavitation plate over the outlet of the reservoir. This is a common piece of sheet metal at least 1/8 inch thick set over and above the outlet opening. This plate will allow flow into the outlet from a horizontal direction and effectively extends and enlarges the reservoir opening. This prevents the vortex condition from developing.

F. Release of Air Suspended within the Fluid

There is considerable air suspended in cold hydraulic fluid. As the fluid warms, air is released into the system. A reduction of fluid pressure will also release air out of suspension. A simple relief valve poppet can create an orifice that increases velocity of the fluid and lowers its pressure. The reduced pressure condition releases air out of suspension into the system. Relief valves should be returned below the fluid level of the reservoir as far from the reservoir outlet as possible. This allows time for the air released by the relief valve to be removed before leaving the reservoir and entering the inlet area of the pump.

In some cases, special return line configurations are needed, or air bleed valves used, to remove air from the system.

A special baffle made of 60 mesh screen can be installed into the reservoir. This baffle should be positioned at a 30° angle in the reservoir so that inlet oil is above the screen and outlet oil is below the screen. The top of the screen should be below the reservoir fluid level far enough to prevent surface foam from coming in contact with the screen. Surface foam can penetrate through the screen into the outlet area. The screen baffle will eliminate all bubbles except the very small ones from the fluid if designed properly.


BETA RATIO VERSUS EFFICIENCY
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BETA RATIO

GRAPH 4

VICKERS PRODUCTS	ISO CLEANLINESS CODE REQUIRED	OLD "SAE CLASS CODE"
Vane Products	18/15	6
Piston Products	16/13	4
All except PVA120	16/12	
Pressure Controls	18/15	6
Proportional Pressure Controls	16/13	4
Flow Controls	18/15	6
Proportional Flow Controls	16/13	4
Directional Controls	18/15	6
Proportional Directional Ctls.	16/13	4
Servo Valves	15/11	2-3
Hydraulic Motors		
Piston	16/13	4
Vane	18/15	6
HMA	18/15	6
Gear Products	16/13	4

To obtain acceptable, long service life of these products when used in industrial applications, provide full flow filtration using filters selected from Vickers series OFP, OFR and OFRS.

Consult your Vickers sales representative for recommended ISO cleanliness codes when these products are used in mobile applications.

CONTENTS

CHAPTER 1	Introduction.	Page 2
CHAPTER 2	Dirt input. Types and sources of contamination.	Page 2
CHAPTER 3	Effects of types and sizes of particles.	Page 3
CHAPTER 4	Specifying contamination levels.	Page 7
CHAPTER 5	Selecting the filter.	Page 8
CHAPTER 6	Locating the filter.	Page 15
CHAPTER 7	Design steps and worked examples.	Page 19
CHAPTER 8	Fluid sampling and analysis.	Page 23
CHAPTER 9	Conclusion.	Page 27

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CHAPTER 1
Introduction

The selection of a filter and its proper location in a hydraulic system needs as much care and expertise as the selection of other components such as pumps, valves and cylinders. Many system designers look no further than the filter manufacturer's catalog, with little or no regard for the particular system requirements. The parameters used to select a filter are usually pressure requirements, flow capability and nominal micrometre rating. This will ensure that the filter does not burst or leak due to excessive pressure and that recommended flow velocities and pressure drops are not exceeded. However, without considering filtration efficiencies or dirt holding capacities, it does not ensure that the system contamination is maintained at acceptable levels.

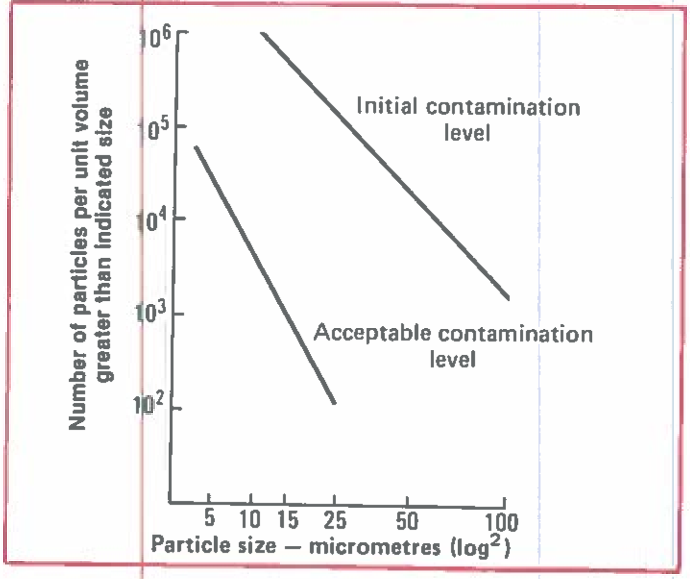
Admittedly, the hit-or-miss approach described above often appears to achieve an acceptable result, but with over 70% of hydraulic system failures known to be due to poor fluid condition, there is a clear need for a more systematic approach to contamination control.

A Systematic Approach to Filtration

To work towards the most effective protection consistent with economy, we must first define our goal. It is not, as is widely assumed, to simply separate out particles larger than a certain size chosen arbitrarily. Instead, we must achieve stable levels of contamination acceptable and appropriate to various parts of the system.

For a stabilized contamination level, 'dirt in' must equal 'dirt out' (collected by the system filters). 'Dirt in' is made up of in-built contaminant, contamination in the initial oil, and contamination drawn in from the atmosphere through the air breather and cylinder seals — all of which contribute to the generation of particles by the process of wear.

Fig. 1: Typical relationship between the desired design contamination level and the actual level prior to start-up. It is essential that the system is flushed and run at no-load until the acceptable contamination level is achieved.



The effect of the inbuilt contamination must be carefully considered. Inevitably, it will be high even when care is taken in the preparation of pipework and manifold blocks. Flushing will remove some initial contaminant, but there are many systems where this is not done and, at the first start-up, high pressures are generated with high contamination levels present. The result is usually rapid pump wear and valve malfunction, the first of which will almost certainly go undetected at this time. *All hydraulic systems should be run in an off-load condition until the desired contamination level is attained.*

Fig. 1 shows a typical relationship between the design contamination level and the actual level prior to start-up. This convenient method of presentation follows naturally from the logarithmic distribution of particle size that occurs in practice. The relative slopes of the initial and acceptable contamination lines are a clear pointer to the type of filtration needed.

The methods used to determine and control contamination levels will be discussed in more detail later, but at this stage we can summarize the practical and performance requirements of the filtration system as follows:

1) It must be capable of reducing the initial contamination to the desired level within an acceptable period of time, without causing premature wear or damage to the hydraulic components.
2) It must be capable of achieving and maintaining the desired level, and allow a suitable safety factor to provide for a concentrated ingress which could occur, for example, when a system is 'topped-up'.
3) The quality of maintenance available at the end user location should be considered.
4) Filters must be easily accessible for maintenance.
5) Indication of filter condition to suit the end user's requirements must be provided.
6) In continuous process plants, facilities must be provided to allow changing of elements without interfering with plant operation.
7) The filters must provide sufficient dirt holding capacity for an acceptable interval between element changes.
8) The inclusion of a filter in the system must not produce undesirable effects on the operation of components, e.g., high back pressures on seal drains.
9) Sampling points must be provided to monitor initial and subsequent levels of contamination.

CHAPTER 2
Dirt Input. Types and Sources of Contamination

New Oil

Although oil is refined and blended under relatively clean conditions, it is usually stored in drums or in a bulk tank at the user's factory. At this point it is no longer clean because the filling lines contribute metal and rubber particles, and the drum always adds flakes of metal or scale. Storage tanks can be a real problem because water condenses in them to cause rusting, and contamination from the atmosphere finds its way in unless satisfactory air breather filters are fitted.

If the oil is being stored under reasonable conditions, the principal contaminants on delivery to a machine will be metal, silica and fibres. With oils from reputable suppliers, sampling has shown average counts of 30,000 to 50,000 particles above 5 micrometre per 100 ml, with a relatively low silt level. Using a portable transfer unit or some other filtration arrangement, it is possible to remove much of the contamination present in new oil before it enters the system and is ground down into finer particles.

Built-in Contamination

New machinery always has a certain amount of built-in contamination. Care in assembly and in flushing the system reduces this but never eliminates it. Typical built-in contaminants are burrs, chips, flash, dirt, dust, fibre, sand, moisture, pipe sealant, weld splatter, paints and flushing solutions.

The amount of contaminant removed during flushing depends not only on the effectiveness of the filter used, but also the velocity of the flushing fluid. Unless high velocities are attained, much of the contaminant will not be dislodged until the system is in operation, with component failure the almost certain result. *Irrespective of the standard of flushing, an off-load running-in period should be regarded as essential.* Some built-in contaminant such as weld scale often remains intact until high pressure oil is forced between it and the parent metal, thus loosening it.

Environmental Contamination

Contaminants from the immediate surroundings can be introduced into a fluid power system. On large installations, such as those within steelworks, it is relatively easy to ascertain the environmental conditions, though they vary considerably. For example, a coke oven system operates in conditions very different from a cold mill. Sometimes the best solution is to protect the hydraulic equipment by providing a clean room where maintenance can be carried out under controlled conditions. Unfortunately, it is not uncommon to see hydraulic power sources exposed to the worst possible environment, while alongside, the electronic controls are protected by pressurized and temperature controlled cabins.

In most machine shops the relatively large contaminant particles of 10 - 15 micrometres do not demand a high standard of filtration. On the other hand, foundries and stone quarries demand a very high standard of filtration because of the airborne abrasive particles.

The mobile equipment field presents special problems because the original equipment manufacturer usually sells a standard machine to operate in a wide variety of environmental conditions.

Entry Points for Environmental Contamination

1. Air breathers. Very little information appears to be available on what these filters will actually achieve, and purely nominal ratings are usually specified. The amount of air passing through the filter will depend on draw-off which means, for example, that single acting cylinders operating in bad environments must result in a greater ingress of contaminant.

2. Power Unit Access Plates. In some plants it cannot be assumed that access plates will always be replaced, though this is not as common as it once was. In power unit design good sealing is vital, and in bad environments such items as strainers should not be positioned inside the reservoir if access requires the re-fitting of removable plates. Other removable items will allow ingress during maintenance, and good design practice should minimize this.

3. Cylinder Seals. Wiper seals cannot be 100% effective in removing very fine contaminant from the cylinder rod.

If they were, they would remove the oil film from the piston rod, producing a result that is usually diagnosed as a leaky seal. In any case, a completely dry rod would quickly wear out the seals. Where cylinders remain extended in a heavily contaminated atmosphere, considerable quantities of fine particles can get into the system unless protection such as a bellows is provided.

Dr. E.C. Fitch* has shown that cylinder piston rod seals naturally ingress about six particles over 10 micrometre for each square inch of swept rod area.

Wear of seals or wipers can increase the ingression rate considerably. In bad ambient conditions a 2 inch diameter rod in a 4 inch bore cylinder, cycling at a speed of 36 feet per minute, could ingress in excess of 16,000 particles over 10 micrometres every minute, and this quantity could increase by a factor of 100 for every 100 hours of running. Dr. Fitch quotes these figures as severe conditions, but nevertheless realistic.

Generated Contamination

Contamination is created internally by the operation of a hydraulic system. The generated contaminants are products of wear, corrosion, cavitation and fluid breakdown, i.e. decomposition, oxidation, etc. Experience shows that in a system which has been carefully flushed and has filtered oil added to the sealed reservoir (incorporating an effective breather), the contamination will be mainly system generated.

If the initial level is not satisfactory, this induces wear which greatly accelerates the build up of generated contaminant.

***Dr. E.C. Fitch, Professor and Director of the Fluid Power Research Center, Oklahoma State University.**

CHAPTER 3

Effects of Types and Sizes of Particles

We know that contaminant particles are of all shapes and sizes, and that the finer they are the more difficult it is to count them and to determine the material of which they are composed. However, we can say that the majority are abrasive and that, interacting with surface protrusions, they plough and cut fragments from a surface. This wear accounts for about 90% of failures due to contamination.

Failures arising from contamination fall into three categories:

1. Catastrophic Failure occurs when a large particle enters a pump or valve. For instance, if a particle causes a vane to jam in a rotor slot, the result may well be complete seizure of the pump or motor. In a spool type valve, a large particle trapped at the right place can stop a spool closing completely. Another example of catastrophic failure occurs when the pilot orifice of a valve is blocked by a large particle. Fine particles can also cause catastrophic failure; a valve, for example, can fail to operate due to silting.

2. Intermittent Failure is caused by contaminant on the seat of a poppet valve which prevents it from reseating properly. If the seat is too hard to allow the particle to be embedded into it, the particle may be washed away when the valve is opened again. Later, another particle may prevent complete closure only to be washed away when the valve opens. Thus a very annoying type of intermittent failure occurs.

3. Degradation Failure follows wear, corrosion and cavitation erosion. They cause increased internal leakage in the system components, but this condition is often difficult to detect.

The eventual result, particularly with pumps, is likely to be catastrophic failure. The particles most likely to cause wear are clearance-size particles which just pass through the clearances between moving parts (Fig. 2).

Manufacturing clearances within hydraulic components can be divided into two principal zones, i.e. around 5 micrometres for high pressure units, and 15-40 micrometres for low pressure units. The actual clearance may vary considerably, depending on the type of unit and operating conditions it sees. Good component design is important to minimize the effect of small clearances.

We should therefore look at the factors affecting critical clearances and also at the type of failure occuring in various groups of components.

Pumps

All hydraulic pumps have component parts which move relative to one another, separated by a small oil filled clearance. Generally these components are loaded toward each other by forces related to pressure, and the pressure always tends to force fluid through this clearance.

As the life of most pumps is determined by a very small quantity of material being removed from a few surfaces, it follows that if the fluid within the clearance is heavily contaminated, rapid degradation and eventual seizure could occur. With low pressure units, the design permits relatively large clearances and contamination has less effect. Also at the lower pressure, there is less force available to drive particles into critical clearances. Increasing the pressure, therefore, is of major significance in determining the effect of contamination on a pump.

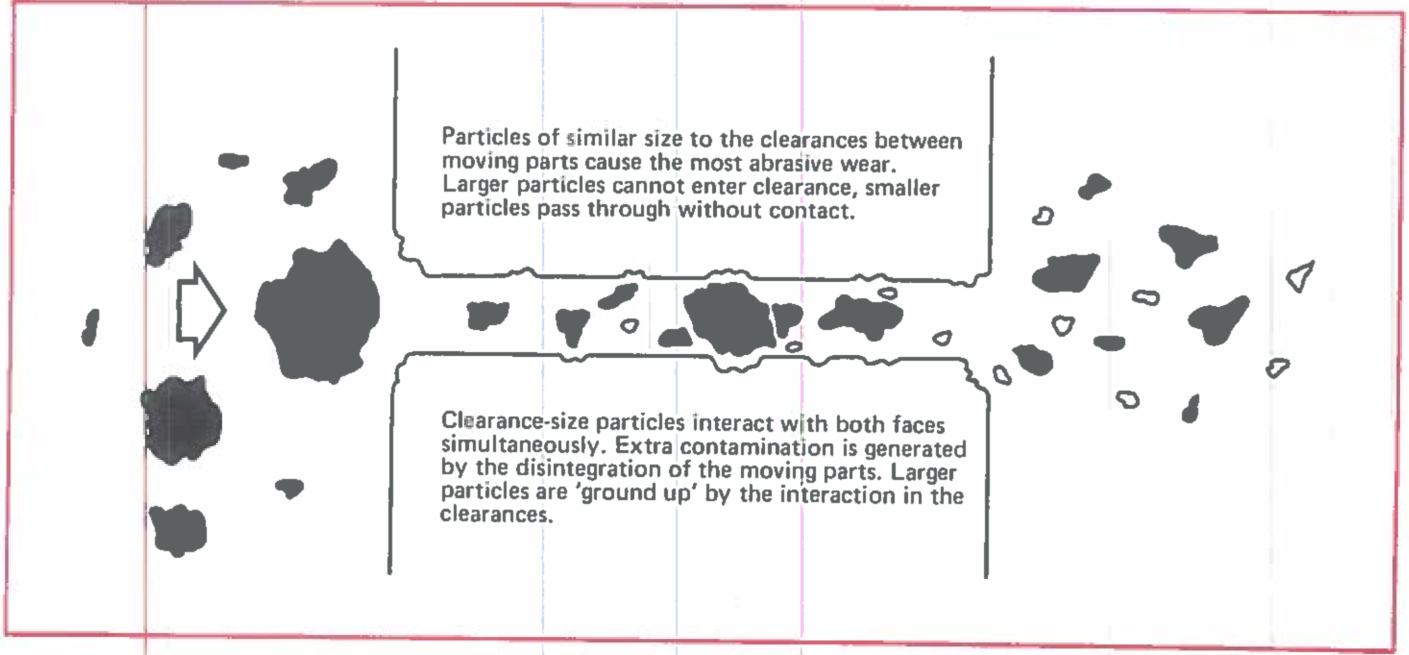
Another factor affecting clearances is the oil film thickness, which is also related to fluid viscosity. An optimum viscosity is used for design, which provides good film thickness to support loads hydrodynamically, but which is also low enough to allow adequate filling of the pump without cavitation. It is generally found in practice that filtration requirements become less critical where higher viscosities are used, and for this reason the maximum viscosity which is compatible with the inlet conditions should be chosen. Similarly, good temperature control will be of benefit in this respect.

The areas in pumps particularly subject to these clearance problems are:

Vane pump — Vane tip to cam ring, rotor to side plate, vane to vane slot.
Gear pump — Tooth to housing, gear to side plate.
Axial piston pump — Shoe to swashplate, cylinder block to valve plate, piston to cylinder block.

Fig. 3, 4 and 5 illustrate the critical areas diagrammatically.

Fig. 2: Interaction of moving parts.



Particles of similar size to the clearances between moving parts cause the most abrasive wear. Larger particles cannot enter clearance, smaller particles pass through without contact.

Clearance-size particles interact with both faces simultaneously. Extra contamination is generated by the disintegration of the moving parts. Larger particles are 'ground up' by the interaction in the clearances.

CHAPTER 7

DIRECTIONAL CONTROLS

Directional valves, as the name implies, are used to control the direction of flow. Though sharing this common function, directional valves vary considerably in construction and operation. They are classified according to their principal characteristics, such as:

- * Type of Internal Valving Element--poppet (piston or ball), rotary spool and sliding spool.
- * Methods of Actuation--cams, plungers, manual lever, mechanical, electric solenoid, hydraulic pressure (pilot-operated) and others, including combinations of these.
- * Number of Flow Paths--two-way, three-way, four-way, etc.
- * Size--nominal size of pipe connections to valve or its mounting plate, or rated gpm flow.
- * Connections--pipe thread, straight thread, flanged, and back-mounted (sometimes called gasket or subplate-mounted).

Finite Positioning

Most industrial directional valves are finite positioning. That is, they control where the oil goes by opening and closing flow paths in definite valve positions. You will note that the graphical symbol for a directional valve will contain a separate envelope (square) for each finite position, showing the flow paths in that position.

CHECK VALVES

A check valve can function as either a directional control or a pressure control. In its simplest form, however, a check valve is nothing more than a one-way directional valve (Fig. 7-1). It permits free flow in one direction and blocks flow in the other.

Note that the composite graphical symbol for a check valve indicates two valve positions--open and closed. This is a rather complicated diagram for such a simple valve, and is seldom used. Rather, the simple ball-and-seat symbol is used universally and therefore will be shown to designate a check valve throughout this manual.

INLINE CHECK VALVES

Inline check valves (Fig. 7-2) are so named because they are connected into the line and the oil flows straight through. The valve body is threaded for pipe or a tubing connector, and is machined inside to form a seat for the poppet or ball (Fig. 7-3). A light spring holds the poppet seated in the normal closed position permitting the valve to be mounted in any attitude.

In the free flow direction, the spring will be overcome and the valve will crack open at about 5 psi pressure drop. The springs are not adjustable, although a variety of sizes are available for special requirements such as creating pilot pressure or as a means of bypassing heat exchangers or oil filters in the event of high flow surges or clogging. In such instances they are not being used as check valves in their true sense but rather as sequence or relief valves.

Although operating pressures of 3000 psi are permissible, the inline check valves are not recommended for applications in which they could be subjected to high velocity return flow.

RIGHT ANGLE CHECK VALVES

A heavier-duty unit, the right angle valve, has a steel poppet and a hardened seat pressed into the iron body (Fig. 7-4 and 7-5). It gets its name from the angle between the flow passage to the poppet and the passage away from the poppet. These valves are built in threaded, flanged connected and back-connected versions. Sizes range from three gpm to 320 gpm with a wide range of cracking pressures.

RESTRICTION CHECK VALVE

A restriction check valve (Fig. 7-6) is a modification of a simple check valve. An orifice plug is placed in the poppet to permit a restricted flow in the normally closed position.

While their usage is somewhat limited, applications would include those which require a free

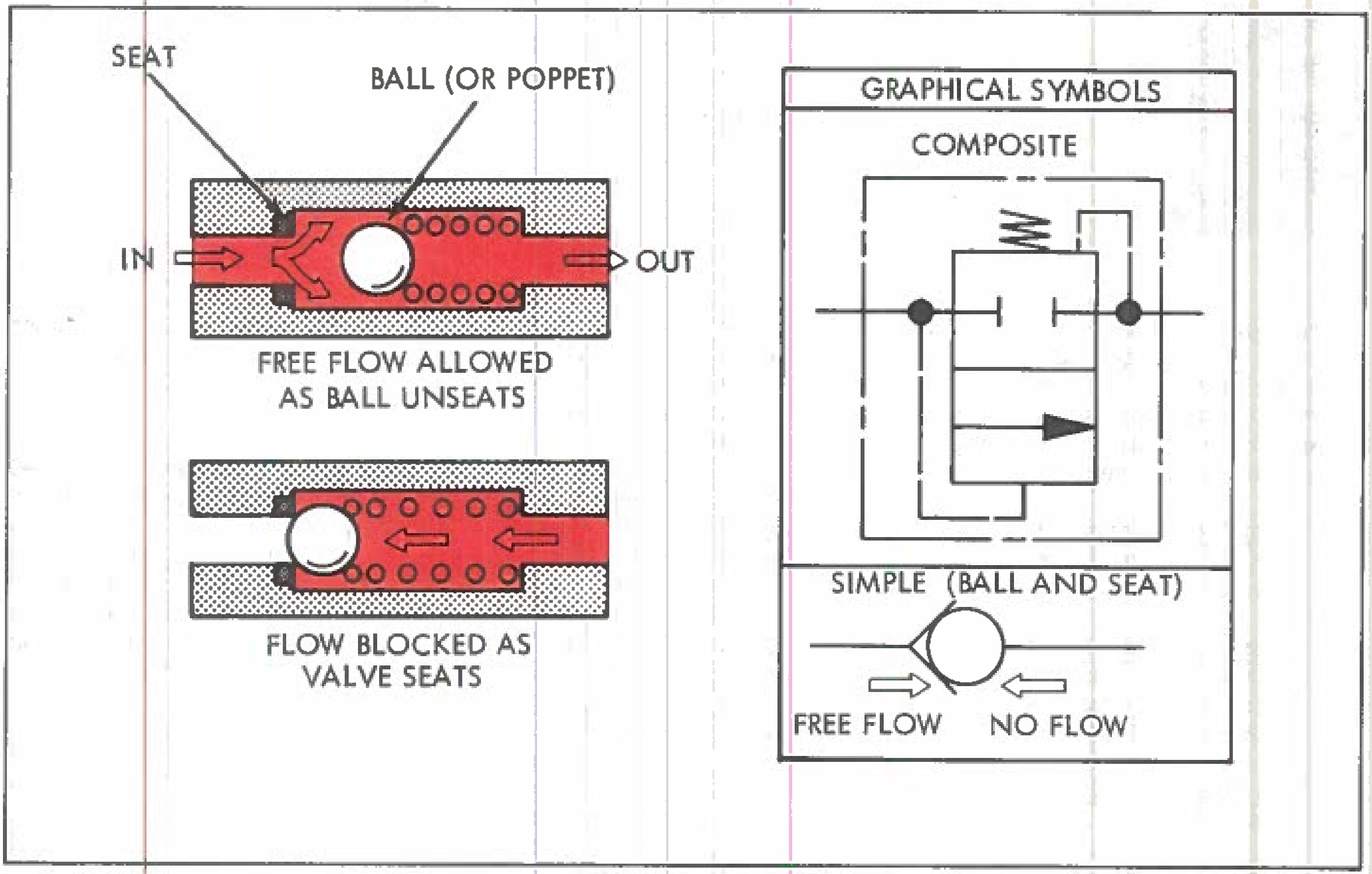


Fig. 7-1. A Check Valve is a One Way Valve

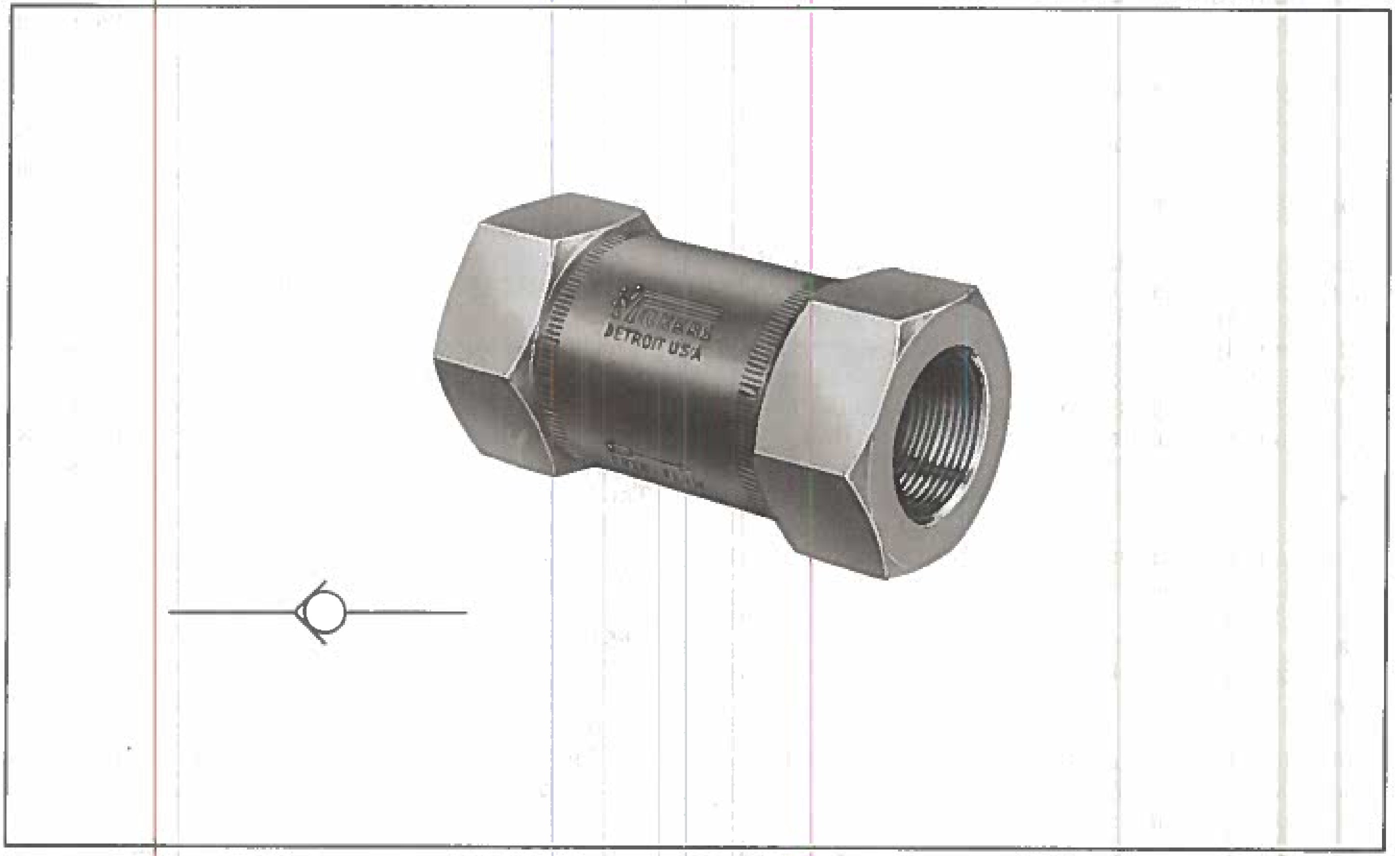


Fig. 7-2 Inline Check Valve

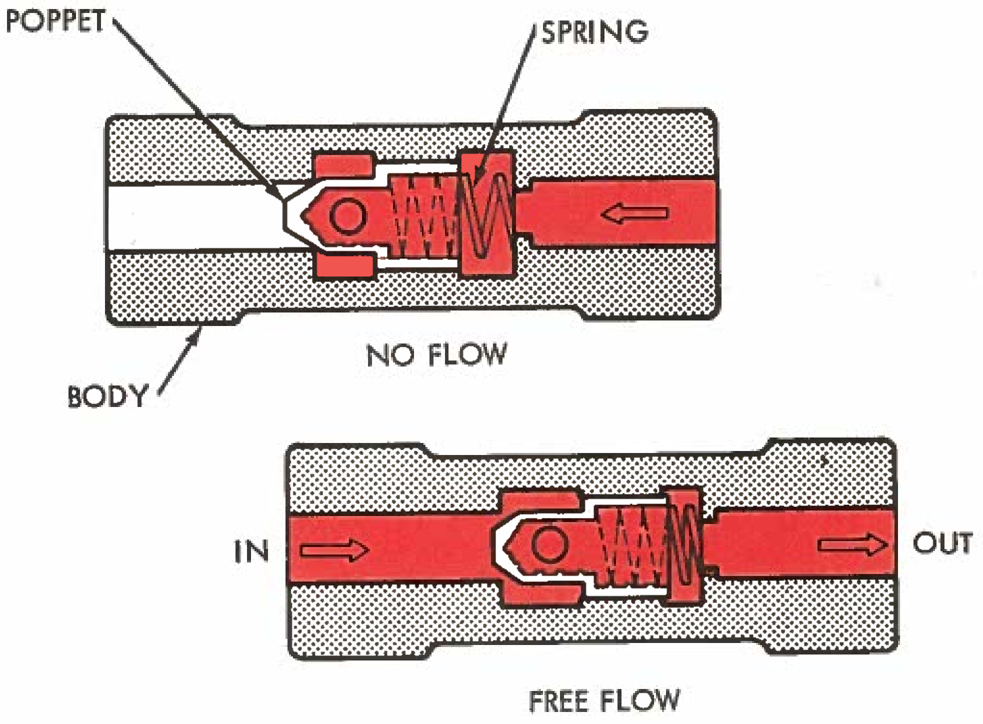


Fig. 7-3. Inline Check Valve Operation

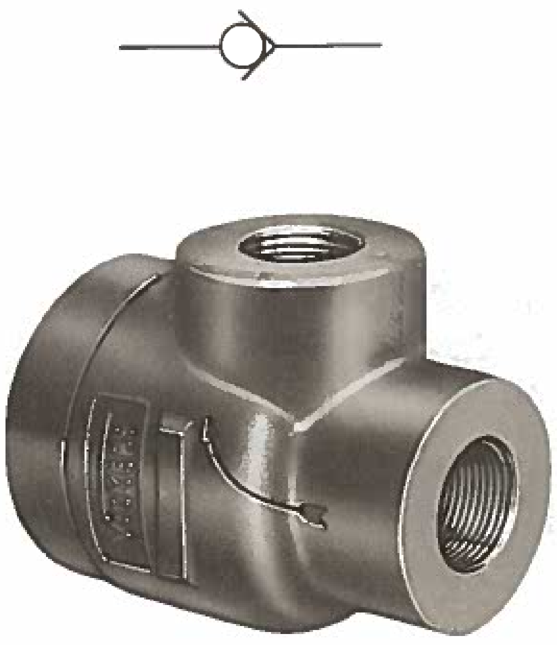


Fig. 7-4. Typical "Right Angle" Check Valve

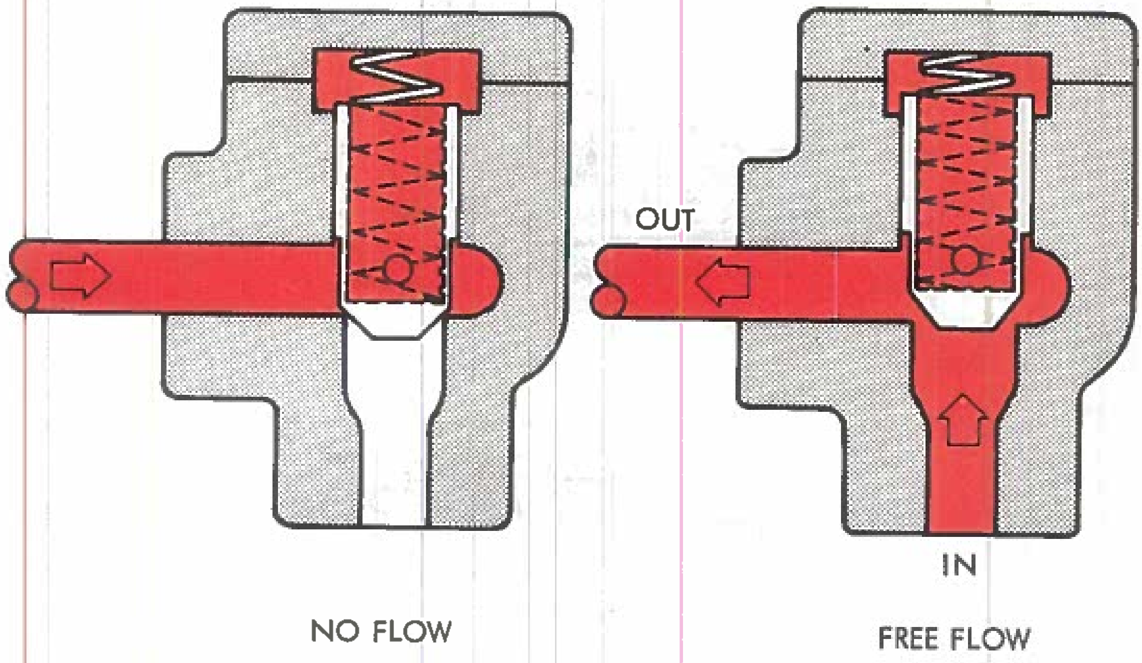


Fig. 7-5. Right Angle Check Valve

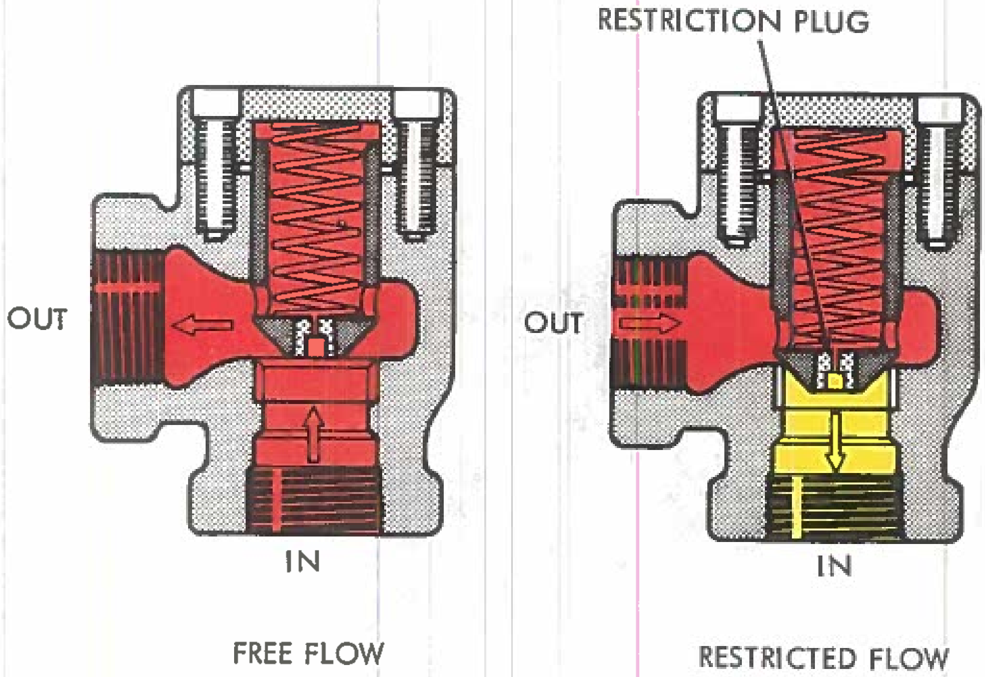


Fig. 7-6. Restriction Check Valve Allows Bleed in No Flow Direction

flow of fluid in one direction and a controlled amount in the other. One example would be in controlling the rate of decompression in a large press.

PILOT-OPERATED CHECK VALVES

Pilot-operated check valves are designed to permit free flow in one direction and to block return flow, until opened by a pressure (pilot) signal. They are used in hydraulic presses as prefill valves--to permit the main ram to fill by gravity during the "fast approach" part of the stroke. They also are used to support vertical pistons which otherwise might drift downward due to leakage past the directional valve spool.

Two designs of pilot-operated check valves are identified as "2C" or "4C" models.

4C Series

Figure 7-7 illustrates the construction of the "4C" type valve. The check valve poppet is lightly spring-loaded against its seat, which is integral with a sleeve that guides the pilot piston. A pilot pressure port in the valve end cover connects to a passage terminating at the head of the pilot piston.

Three operating conditions of the valve are shown in Figure 7-8. In views A and B there is no pilot pressure and the operation is the same as a conventional check valve. In A, pressure at the head of the poppet (valve inlet) has overcome spring force to open the valve and permit flow. In B, pressure is higher on the spring side of the poppet and reverse flow is blocked.

View C shows the condition when pilot pressure is applied to the head of the pilot piston. The stem pushes the poppet off its seat and allows reverse flow. The pilot pressure required to unseat the poppet this way must exceed 40 percent of the pressure in the "outlet" chamber.

2C Series

The "2C" type valve is illustrated in Figure 7-9. In this design, the check valve poppet resembles an automobile engine valve, and has the pilot piston attached to the threaded poppet stem by a nut. The light spring holds the poppet seated in a no-flow condition by pushing against the pilot piston. A separate drain port is provided to prevent oil from creating a pressure buildup on the underside of the piston.

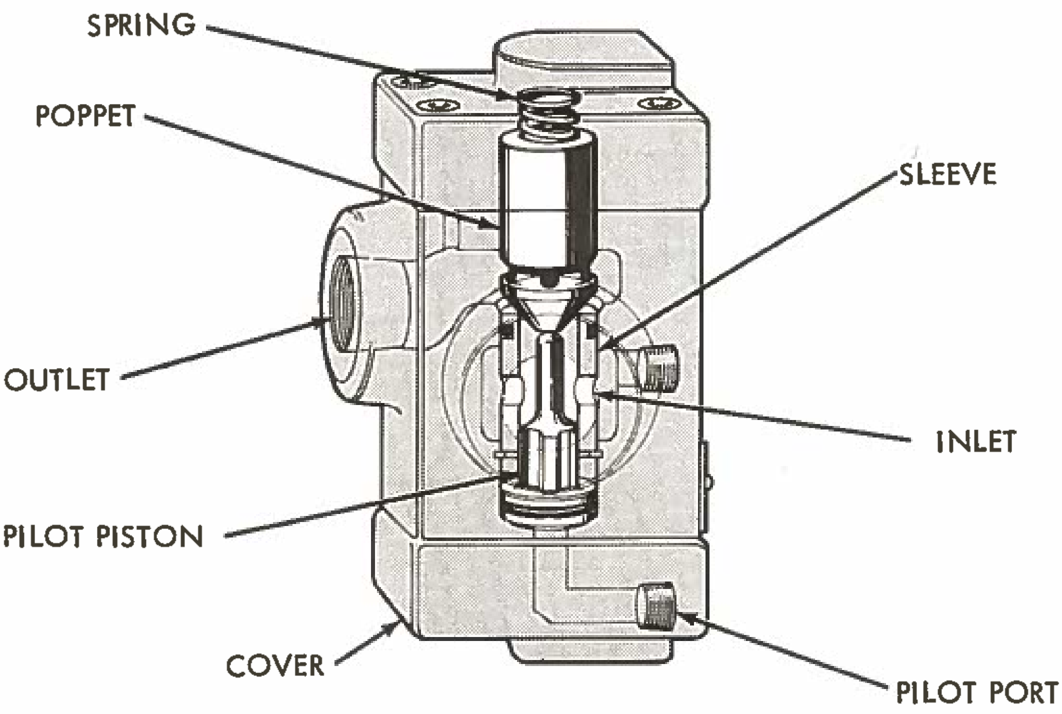


Fig. 7-7. Construction of "4C" Check Valve

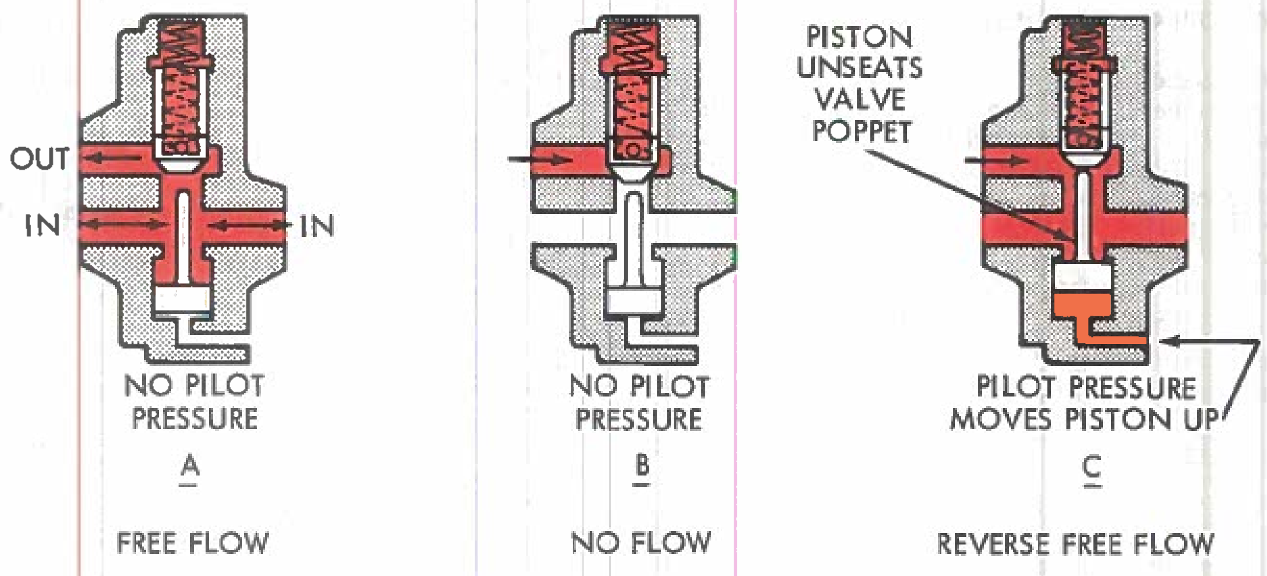


Fig. 7-8. Operation of "4C" Check Valve

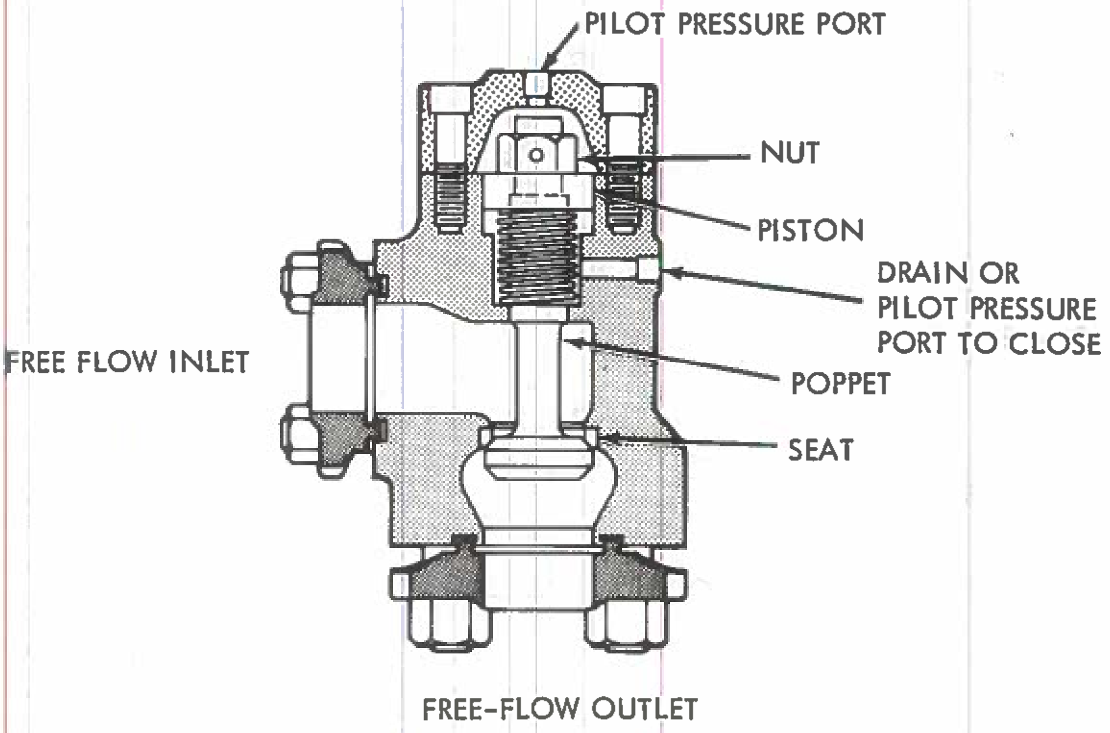


Fig. 7-9. Construction of "2C" Check Valve

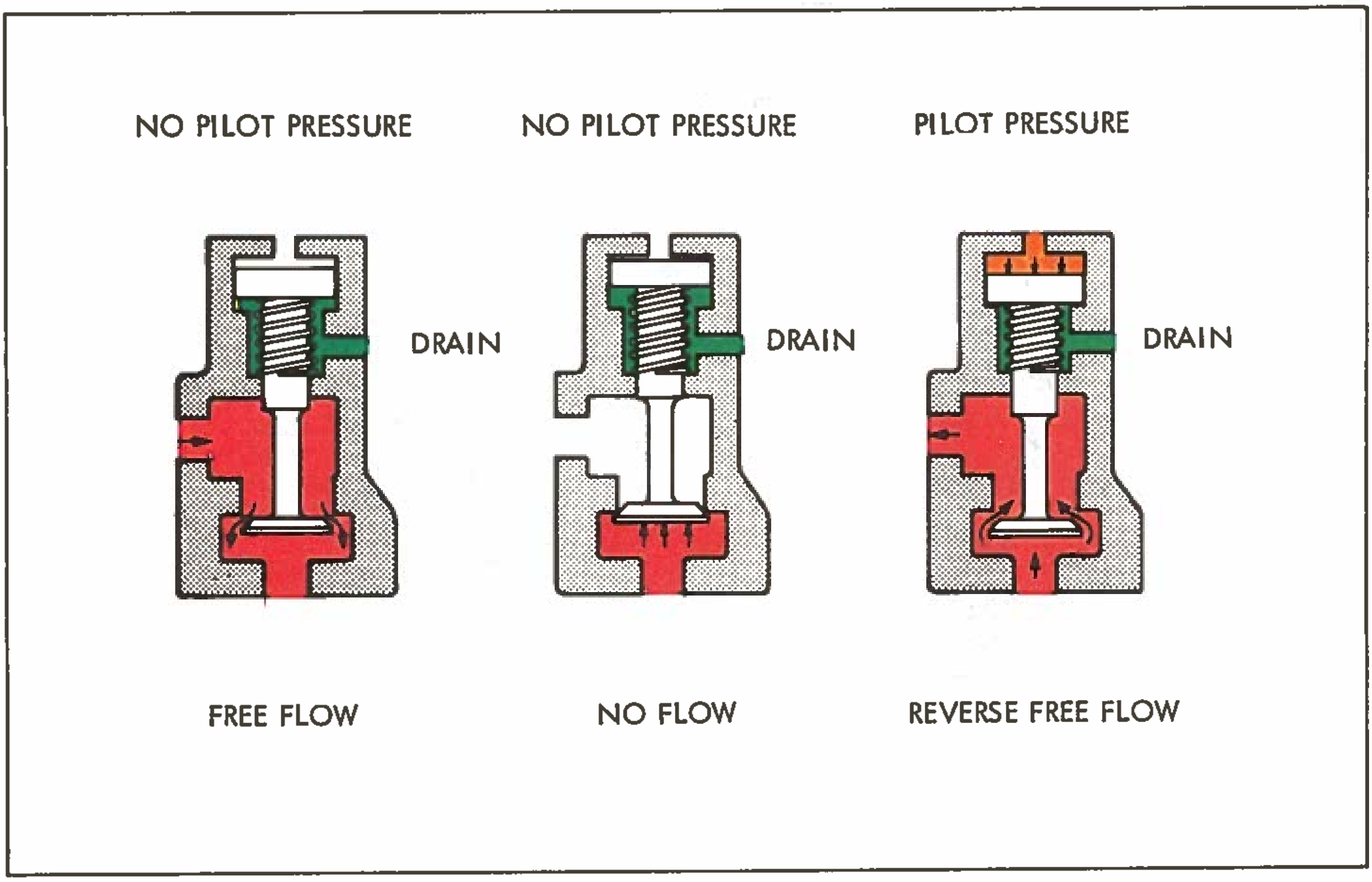


Fig. 7-10. "2C" Check Valve Operation

Figure 7-10, views A and B, show the operation as a conventional check valve with no pilot pressure imposed. Reverse flow (view C) can occur only when a pressure exceeding 80 percent of the pressure in the outlet chamber is effective against the pilot piston.

The valve is also built in a "no-spring" version (Fig. 7-11) for applications where it is desirable to hold the poppet either open or closed. In the no-spring design, the drain and pilot ports both act as pilot pressure ports and are reversed by use of a separate directional valve. Pilot pressure is used to hold the valve in the desired position.

Pilot-Operated Check Valve Applications

Figure 7-12 illustrates the basic operational difference between the "2C" and "4C" valves. In the "4C" type, pressure in the inlet chamber acts against the pilot piston to resist pilot operation. In the "2C" type, inlet pressure assists pilot actuation.

The "4C" valve thus is used in applications where the inlet port is connected to tank during reverse flow. Typical applications are blocking flow around a volume control valve during a feed cycle or preventing a cylinder from drifting due to leakage through a directional valve.

The "2C" valve is used effectively to intermittently block flow from an accumulator. It permits free flow to the accumulator, and can easily be pilot actuated to permit the accumulator to discharge even though pressure is present at both ports.

TWO-WAY AND FOUR-WAY VALVES

The basic function of two-way and four-way valves is to direct inlet flow to either of two outlet ports. As shown in Figure 7-13, flow to the "P" (pump) port of the valve can be directed to either outlet port (labeled A and B for convenience). In the four-way valve the alternate port is open to the tank port permitting return flow to the reservoir. In two-way valves the alternate port is blocked and the tank port serves only to drain leakage from within the valve.

Most of these valves are the sliding spool type, although there are rotary valves which are used principally for pilot control. They are built in two-position or three-position versions. The three-position valve has a center or neutral position. Methods of actuation include manual

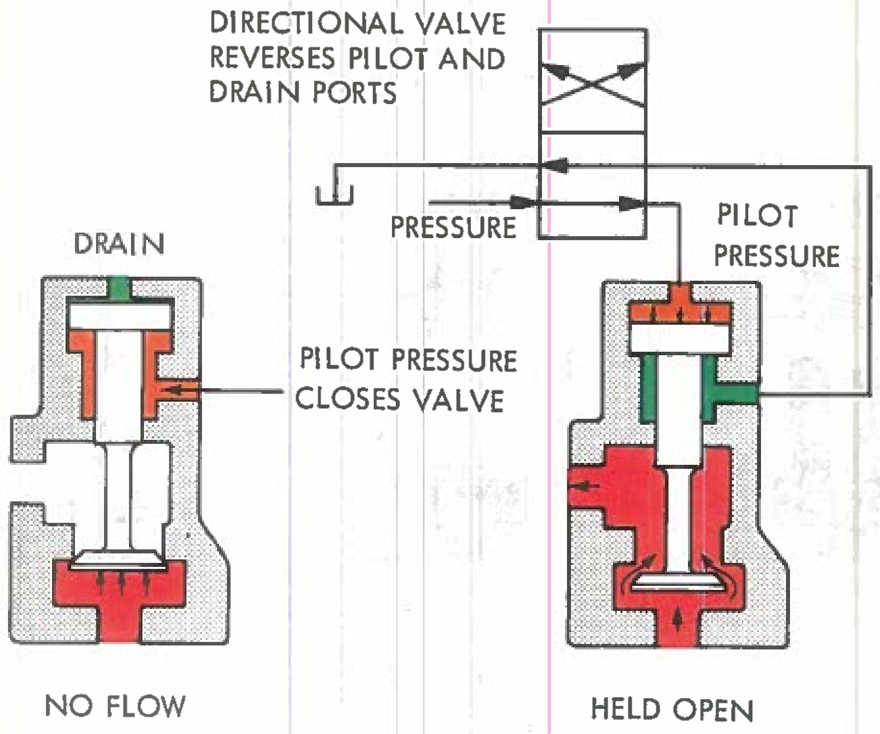


Fig. 7-11. No-Spring "2C" Check Valve

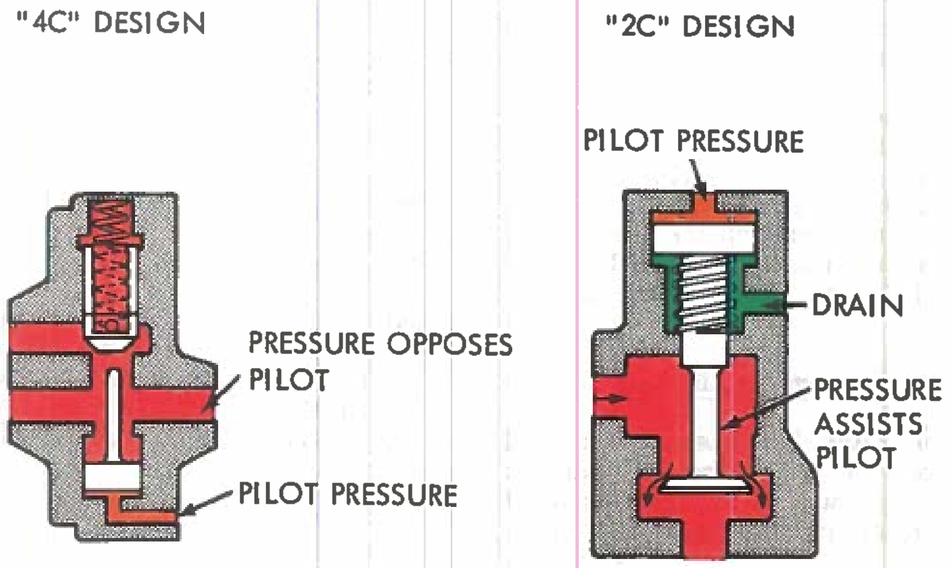


Fig. 7-12. Effects of Pressure on Pilot Control

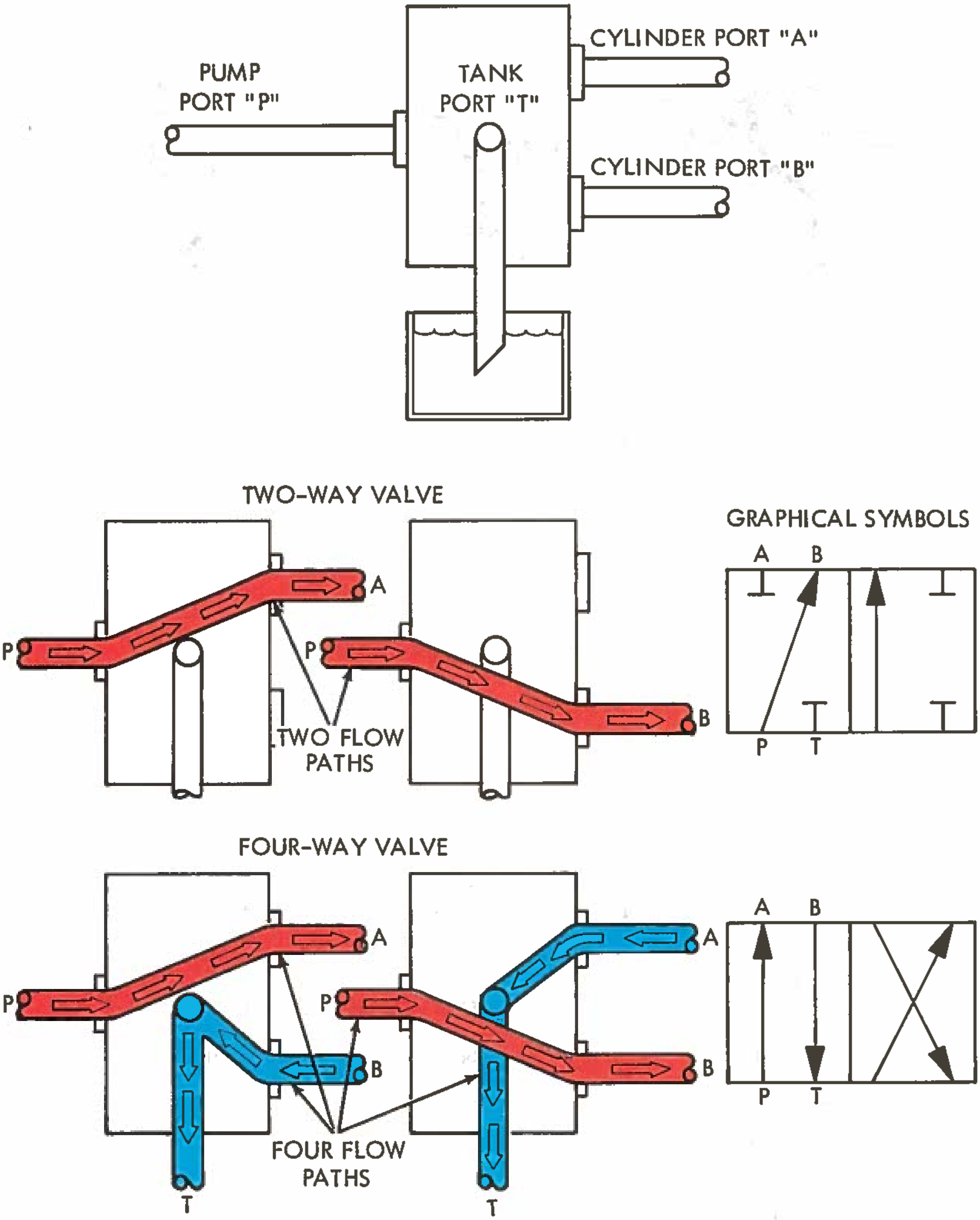


Fig. 7-13. Flow Paths in Two Way and Four-Way Valves

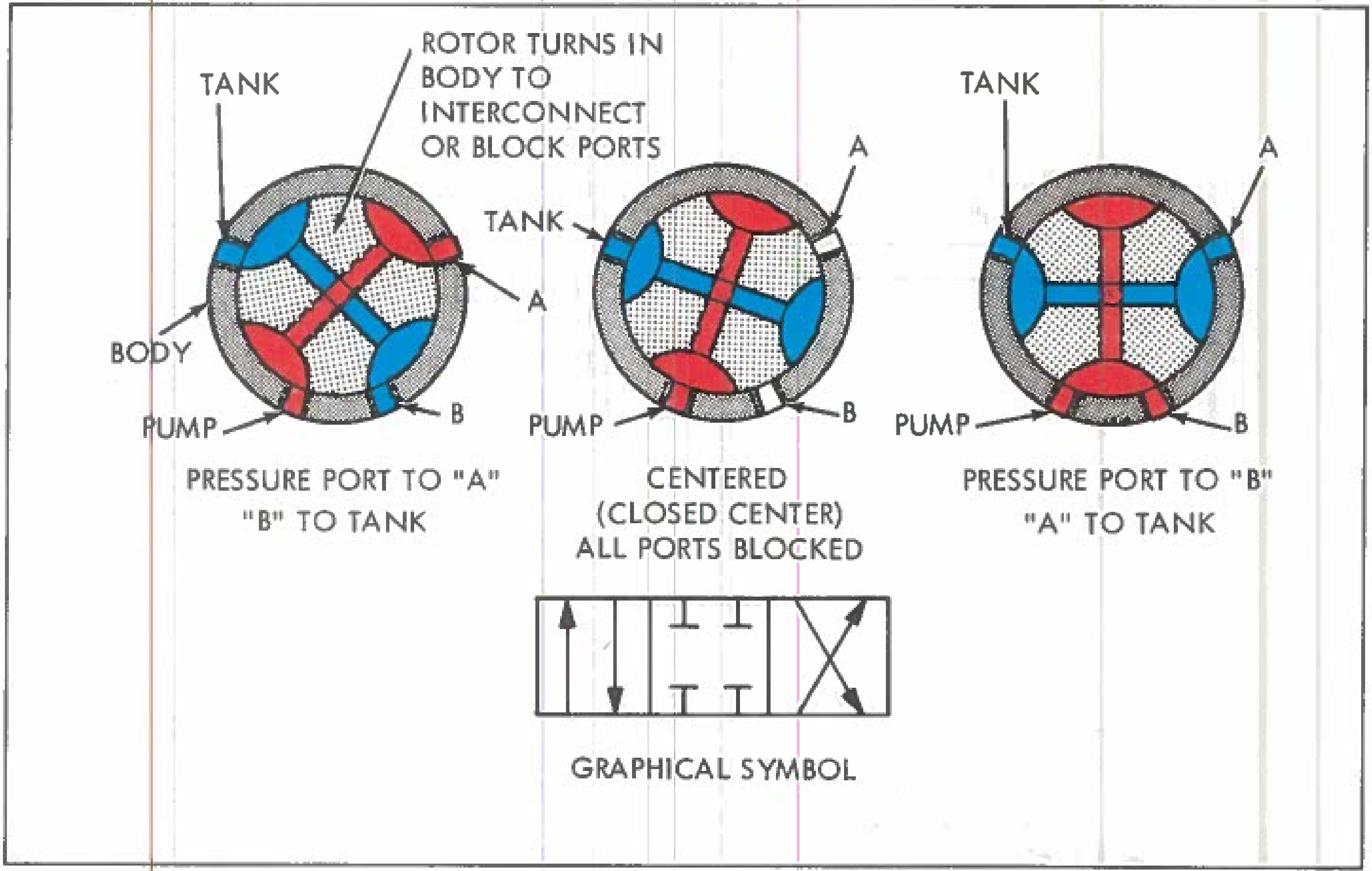

ROTOR TURNS IN BODY TO INTERCONNECT OR BLOCK PORTS
TANK
TANK
TANK
A
A
A
BODY
PUMP
PUMP
PUMP
B
B
B
PRESSURE PORT TO "A" "B" TO TANK
CENTERED (CLOSED CENTER) ALL PORTS BLOCKED
PRESSURE PORT TO "B" "A" TO TANK
GRAPHICAL SYMBOL

Fig. 7-14. Rotary Four-Way Valve

levers, mechanical cams or linkages, springs, solenoids, pilot pressure and others.

ROTARY FOUR-WAY VALVE

A rotary four-way valve (Fig. 7-14) consists simply of a rotor closely fitted in a valve body. Passages in the rotor connect or block the ports in the valve body to provide the four flow paths as shown. A center position can be incorporated if required.

Rotary valves are actuated manually or mechanically. They are capable of reversing cylinders or motors; however, they are used principally as pilot valves to control other valves.

SPOOL TYPE TWO-WAY VALVE

In the spool type directional valve (Fig. 7-15) a cylindrical spool moves back and forth in a machined bore in the valve body. Cored or machined passages from the port connections in the body are interconnected through annular grooves (undercuts) in the spool or blocked by the spool lands.

The two-way version permits selection of two flow paths. In one position, flow is permitted from the "P" port to the "A" port; in the other position from "P" to "B". All other ports and passages are blocked.

SPOOL TYPE FOUR-WAY VALVE

The spool type four-way valve (Fig. 7-16) is identical to the two-way valve in Figure 7-15 except for the machining of the spool lands. The land width is reduced to uncover the "T" (tank) port in the extreme positions and allow return flow to tank.

OPERATING CONTROLS

Spool valves can be actuated or shifted in a number of ways. A typical manually-operated four-way valve is shown with its graphical symbol in Figure 7-16A; a mechanically-operated valve in Figure 7-17. Note that the basic valve symbol is the same, with the addition of the controlling symbol.

Figure 7-18 illustrates a spool type four-way valve that is shifted by air pressure against a piston at either end of the valve spool.

A very common method of actuating a small spool valve is with a solenoid (Fig. 7-19). Elec-

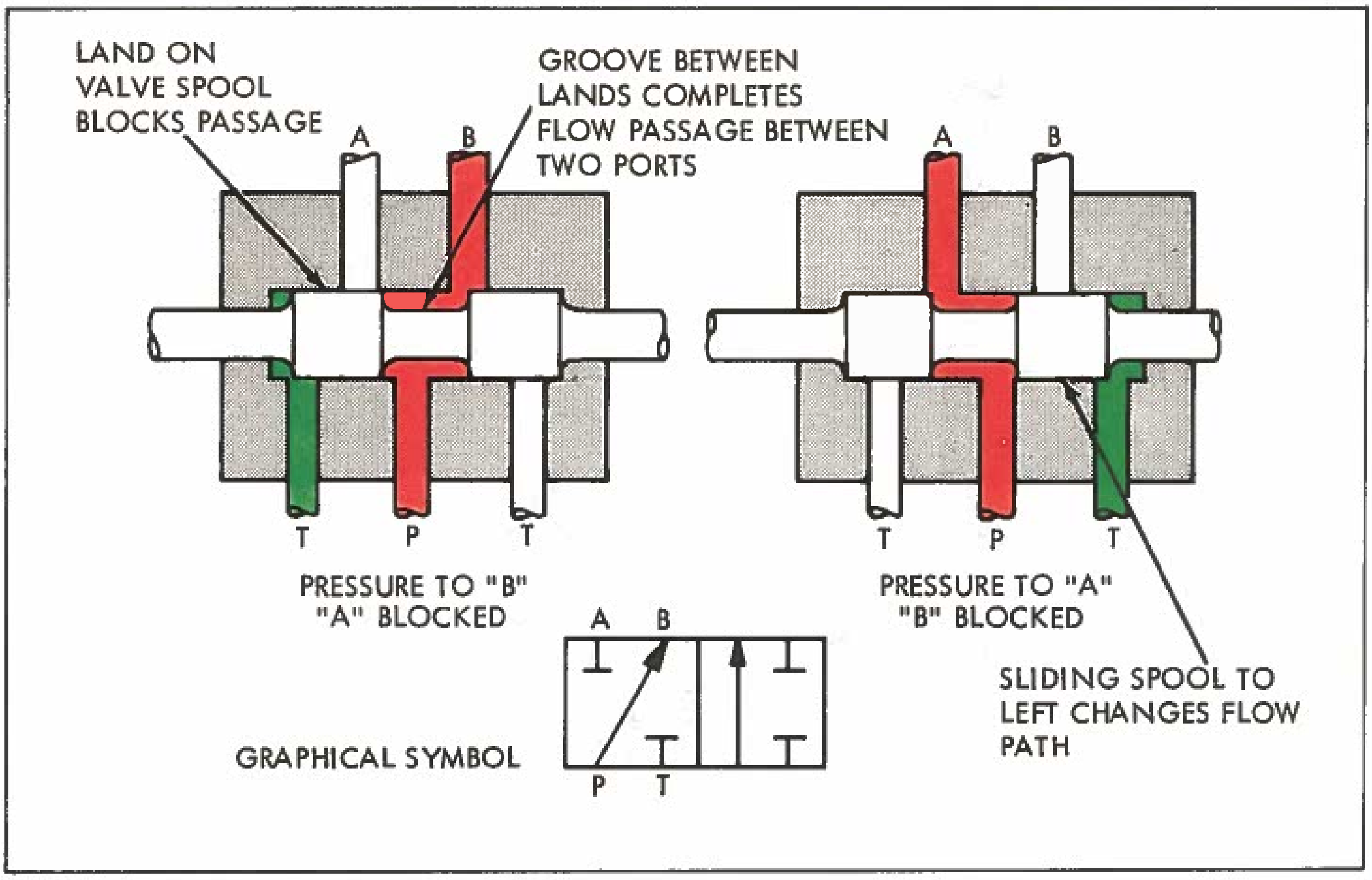

LAND ON VALVE SPOOL BLOCKS PASSAGE
GROOVE BETWEEN LANDS COMPLETES FLOW PASSAGE BETWEEN TWO PORTS
A
B
T
P
T
PRESSURE TO "B"
"A" BLOCKED
A
B
T
P
T
PRESSURE TO "A"
"B" BLOCKED
SLIDING SPOOL TO LEFT CHANGES FLOW PATH
GRAPHICAL SYMBOL
A
B
P
T

Fig. 7-15. Two-Way Spool Valve Slides in Machined Bore

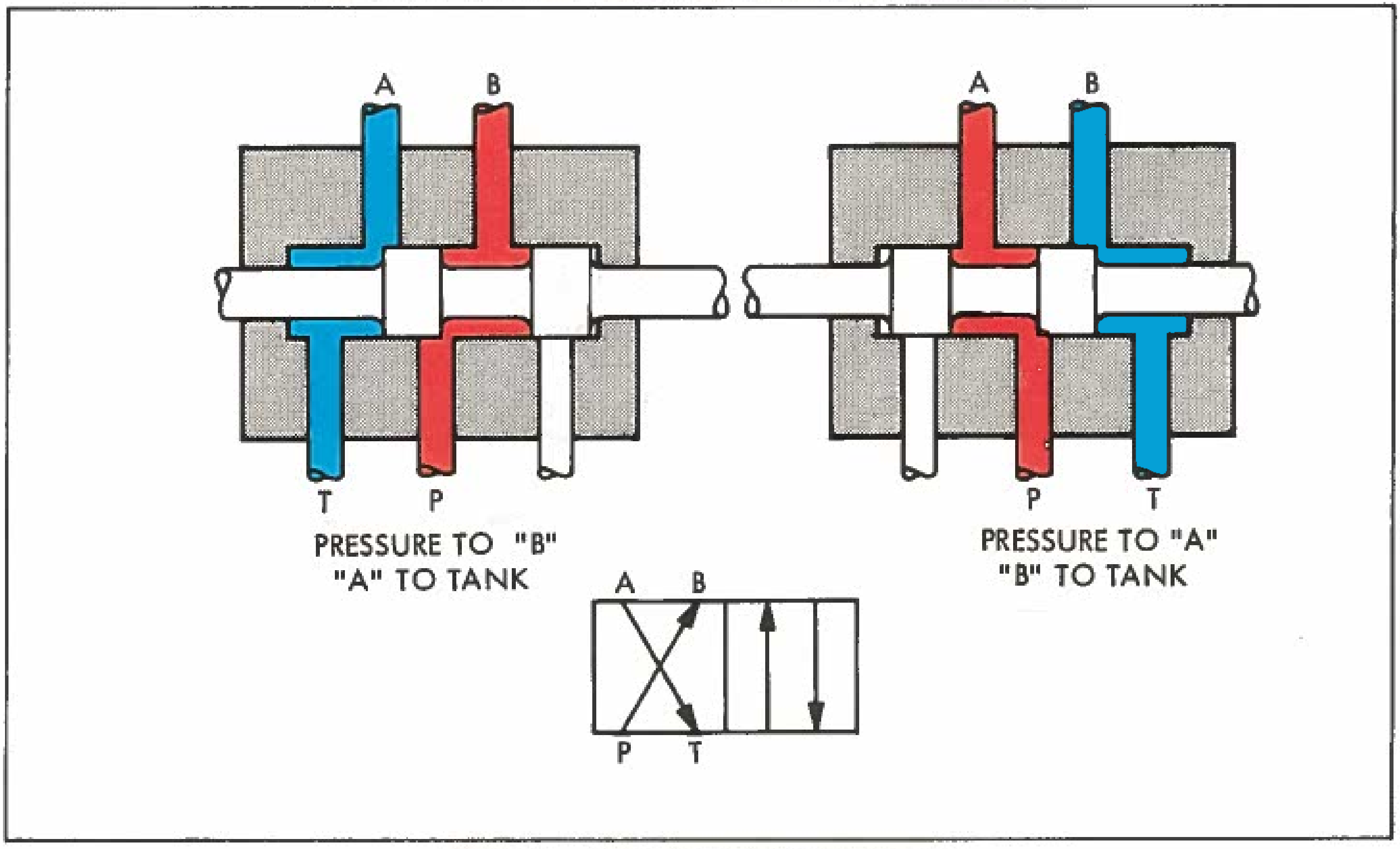


Fig. 7-16. Spool Type Four-Way Valve

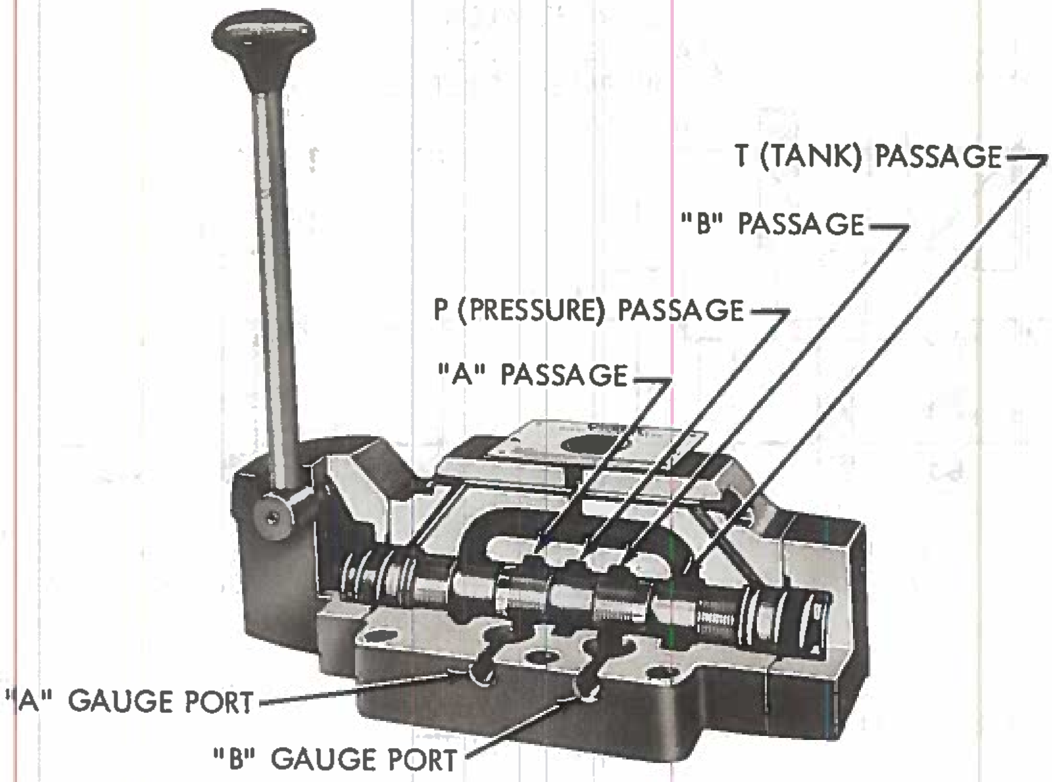


Fig. 7-16A. Manually Operated Four-Way Valve

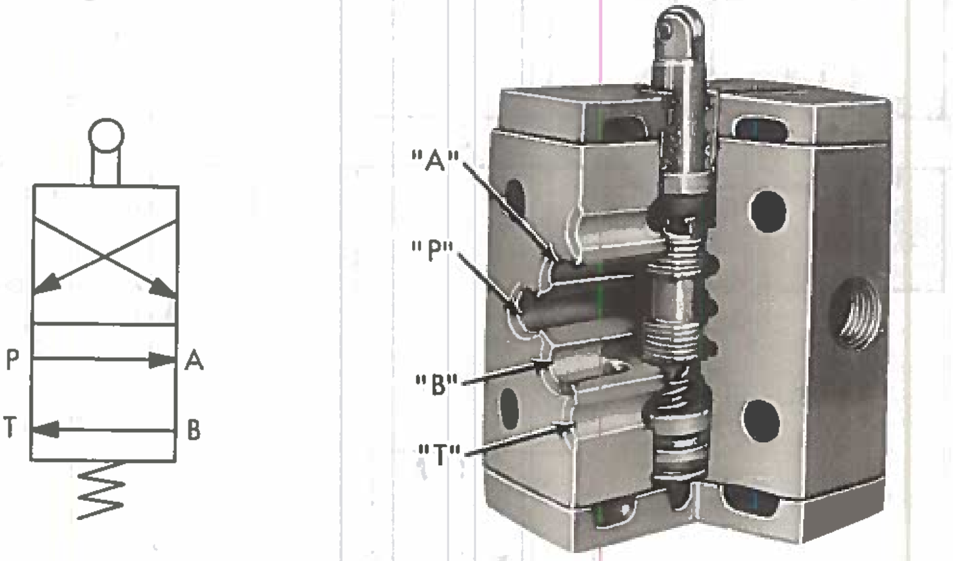


Fig. 7-17. Mechanically Operated Four-Way Valve

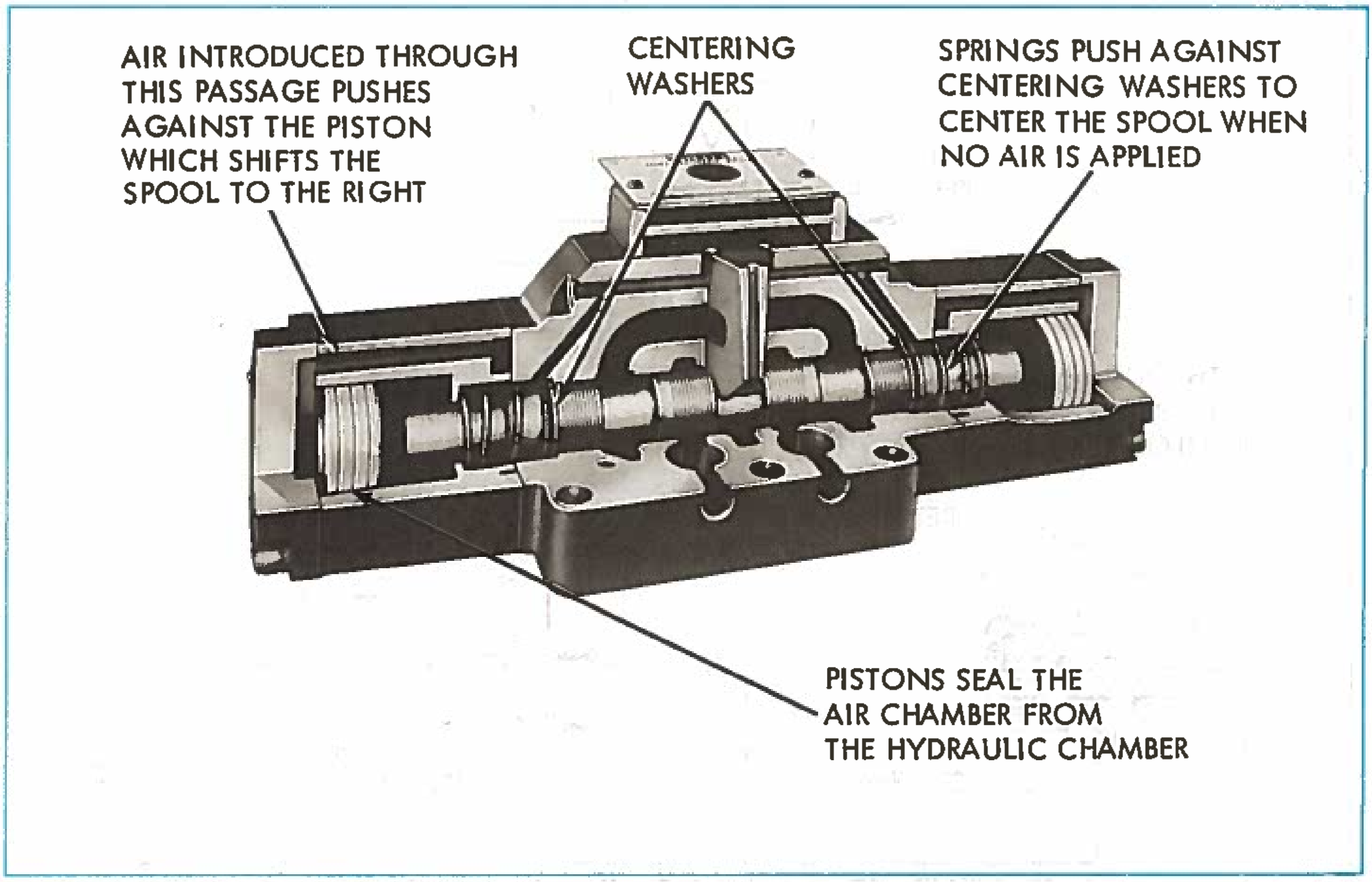

AIR INTRODUCED THROUGH THIS PASSAGE PUSHES AGAINST THE PISTON WHICH SHIFTS THE SPOOL TO THE RIGHT
CENTERING WASHERS
SPRINGS PUSH AGAINST CENTERING WASHERS TO CENTER THE SPOOL WHEN NO AIR IS APPLIED
PISTONS SEAL THE AIR CHAMBER FROM THE HYDRAULIC CHAMBER

Fig. 7-18. Pneumatically-Shifted Four-Way Valve

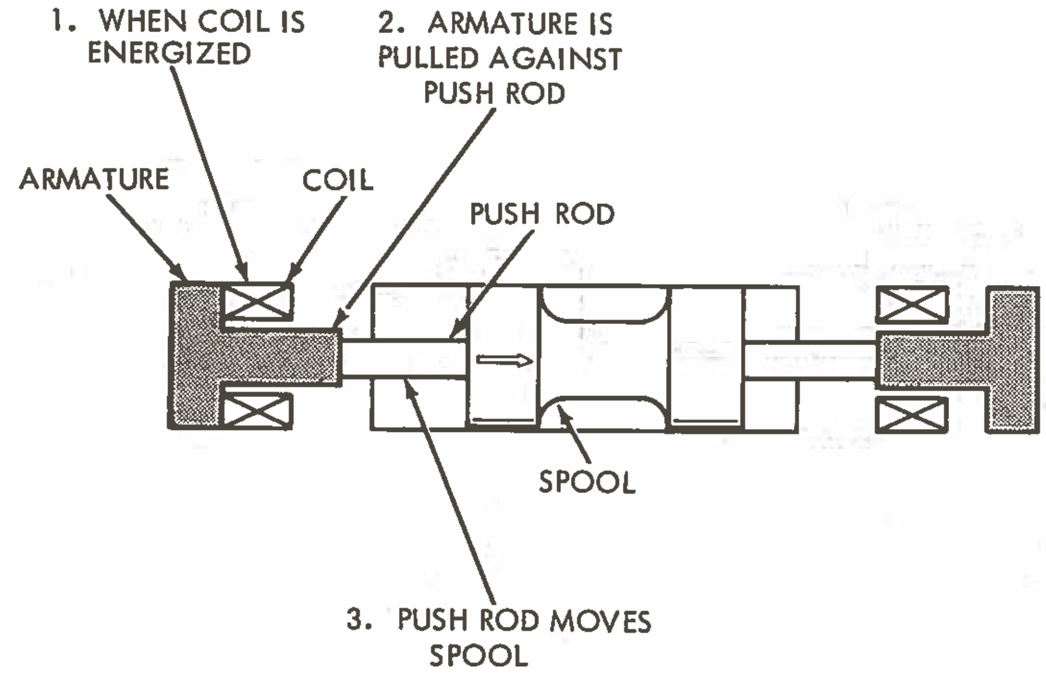

1. WHEN COIL IS ENERGIZED
2. ARMATURE IS PULLED AGAINST PUSH ROD
ARMATURE
COIL
PUSH ROD
SPOOL
3. PUSH ROD MOVES SPOOL

Fig. 7-19. Push-Type Solenoids Shift Many Small Valve Spools

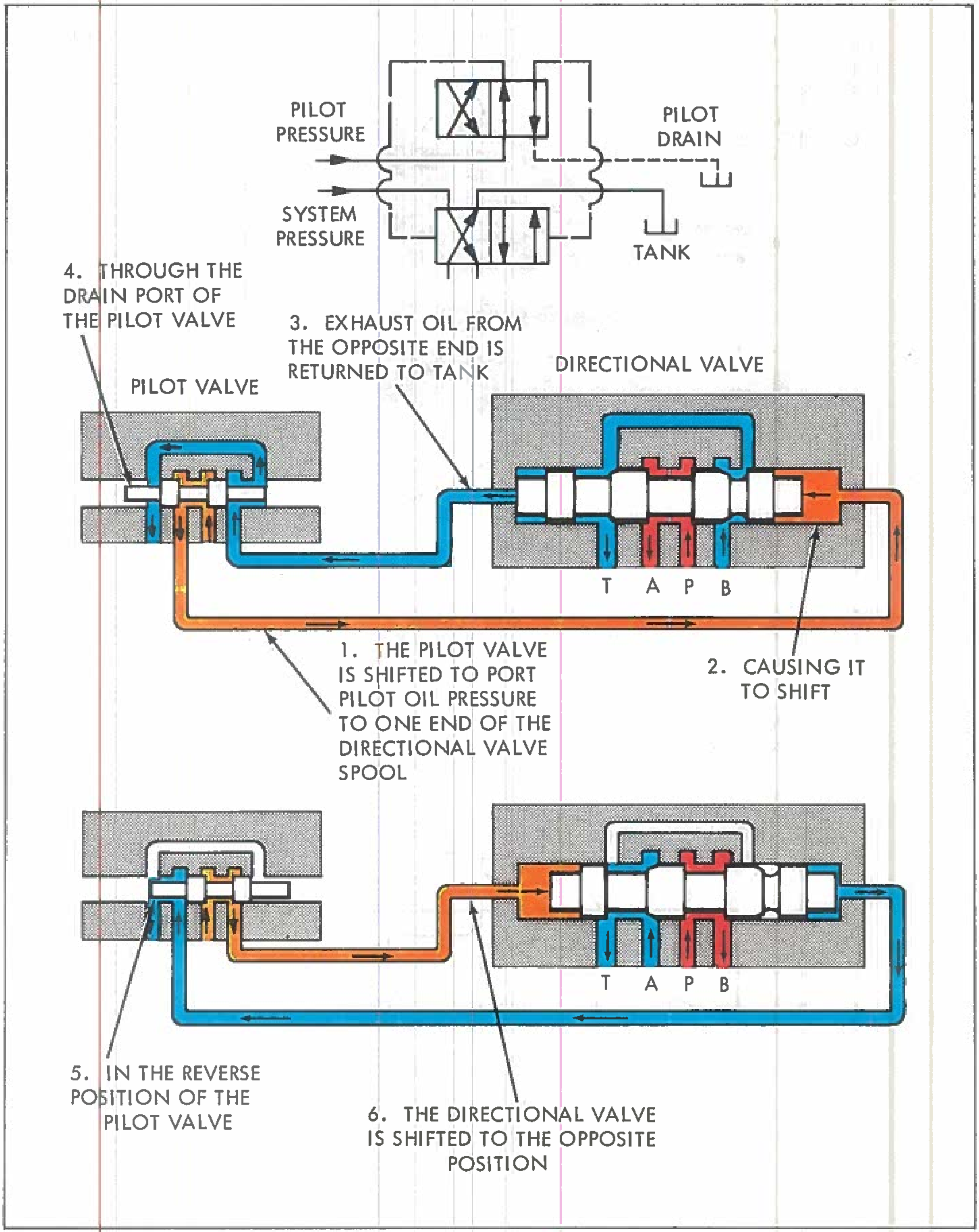

PILOT PRESSURE
SYSTEM PRESSURE
PILOT DRAIN
TANK
4. THROUGH THE DRAIN PORT OF THE PILOT VALVE
3. EXHAUST OIL FROM THE OPPOSITE END IS RETURNED TO TANK
DIRECTIONAL VALVE
PILOT VALVE
T
A
P
B
1. THE PILOT VALVE IS SHIFTED TO PORT PILOT OIL PRESSURE TO ONE END OF THE DIRECTIONAL VALVE SPOOL
2. CAUSING IT TO SHIFT
T
A
P
B
5. IN THE REVERSE POSITION OF THE PILOT VALVE
6. THE DIRECTIONAL VALVE IS SHIFTED TO THE OPPOSITE POSITION

Fig. 7-20. Pilot Pressure is Used To Shift Large Directional Valves

tric energy applied to the solenoid coil creates a magnetic field that draws the armature into the coil. The armature motion is transmitted through a push rod which in turn moves the spool.

In larger valves, the force required to shift the spool is more than it is practical to obtain from a solenoid. Most large directional valves are actuated by pilot pressure against either spool end (Fig. 7-20). The pilot oil is furnished from a smaller four-way valve--referred to as a pilot valve, which while usually actuated by a solenoid may use any of the methods shown in Figures 7-16 through 7-18.

"SPRING CENTERED," "SPRING OFFSET" AND "NO SPRING"

The terms "spring-centered" and "spring-offset" refer to the use of springs returning valve spools to their normal position.

A spring centered valve is returned to the center position by spring force whenever the actuating effort is released. The air-operated valve illustrated in Figure 7-18 is spring-centered.

A spring-offset valve (Fig. 7-21) is a two position valve returned to one extreme position by a spring whenever the actuating effort is released. It is shifted to the opposite position by one of the above methods.

A "no-spring" valve must be actuated entirely by an external control and may "float" between its two positions when the control is released unless it is retained by detents or friction pads. For this reason it is good practice to maintain control on the valve throughout the cycle.

SPOOL CENTER CONDITIONS

Most three-position valves are available with a variety of interchangeable spools. All of the four-way spools provide identical flow patterns in the shifted positions, with different centered conditions as illustrated in Figure 7-22. The open-center type interconnects all ports and the pump delivery can flow to tank at low pressure. The closed center has all ports blocked, so that pump delivery may be used for other operations within the circuit. Otherwise it is forced over the relief valve. Other center conditions permit blocking of selected ports with others open. The tandem type has both cylinder ports blocked in neutral, but the pressure port is open to tank,

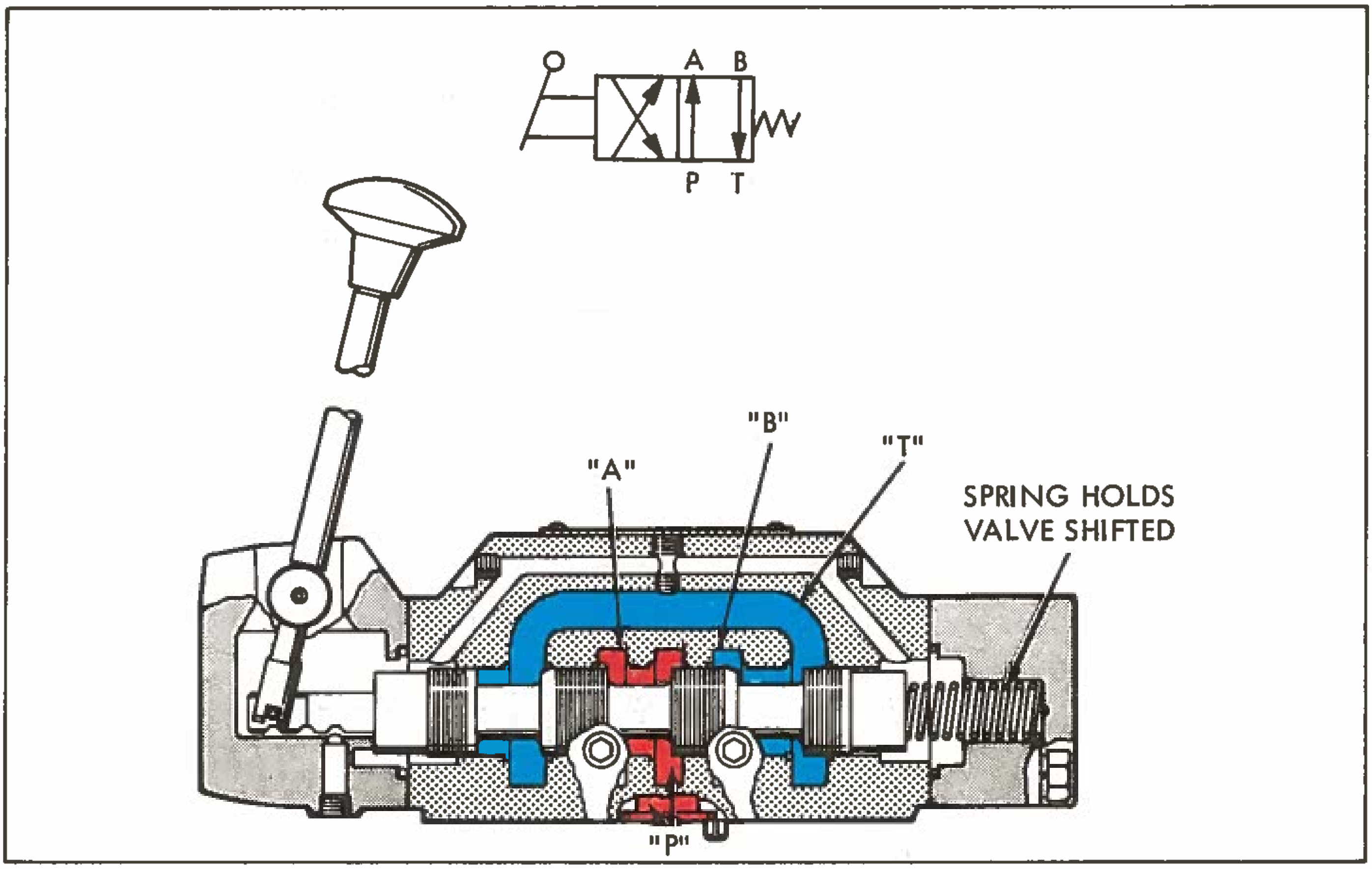


Fig. 7-21. Spring Offset Valve Has Two Positions

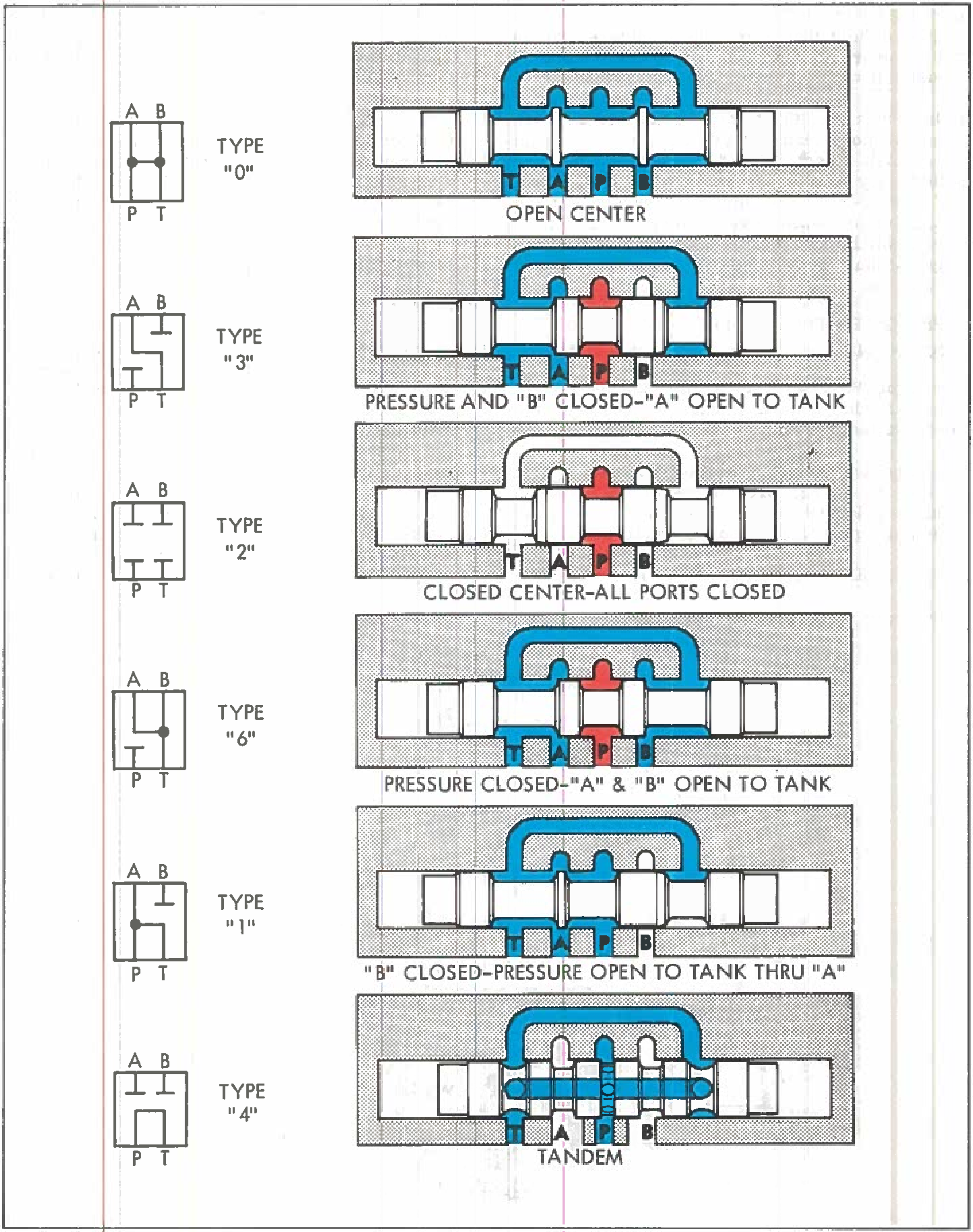
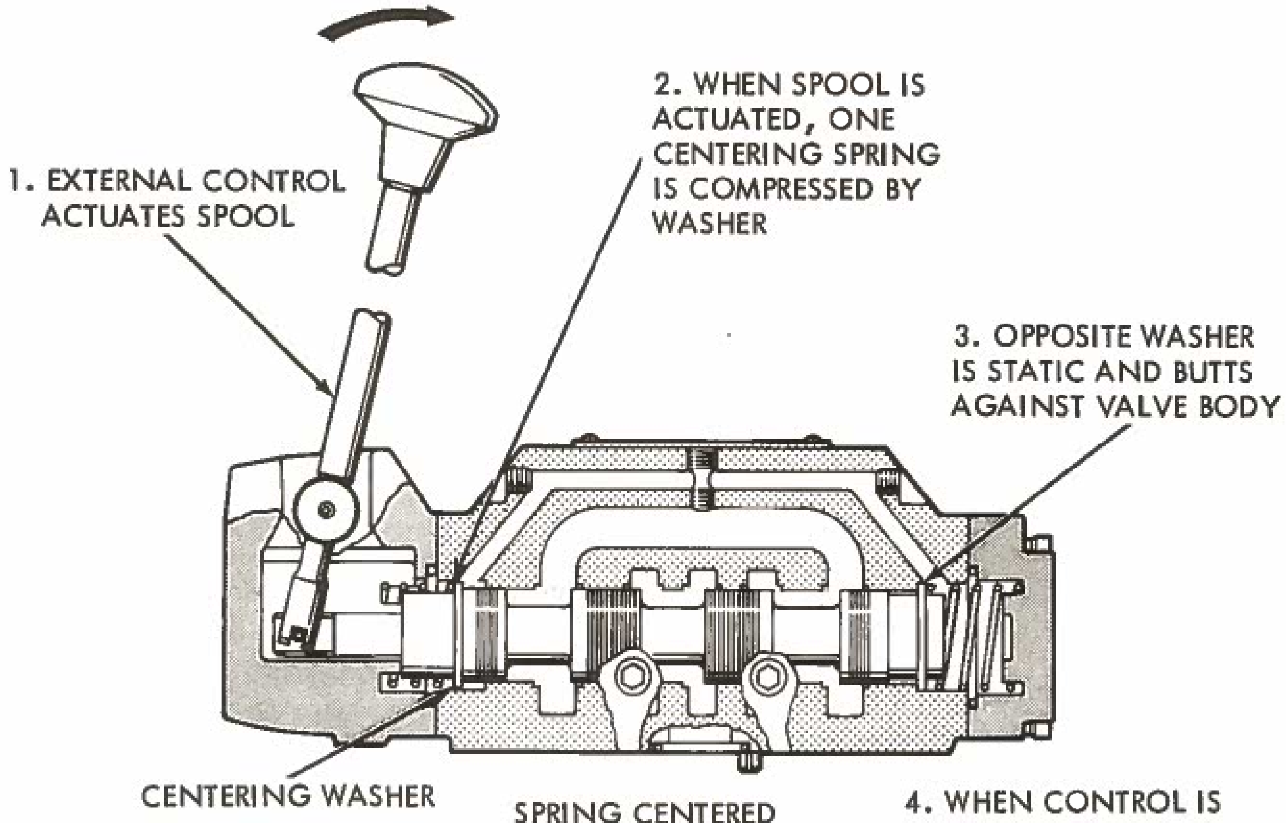
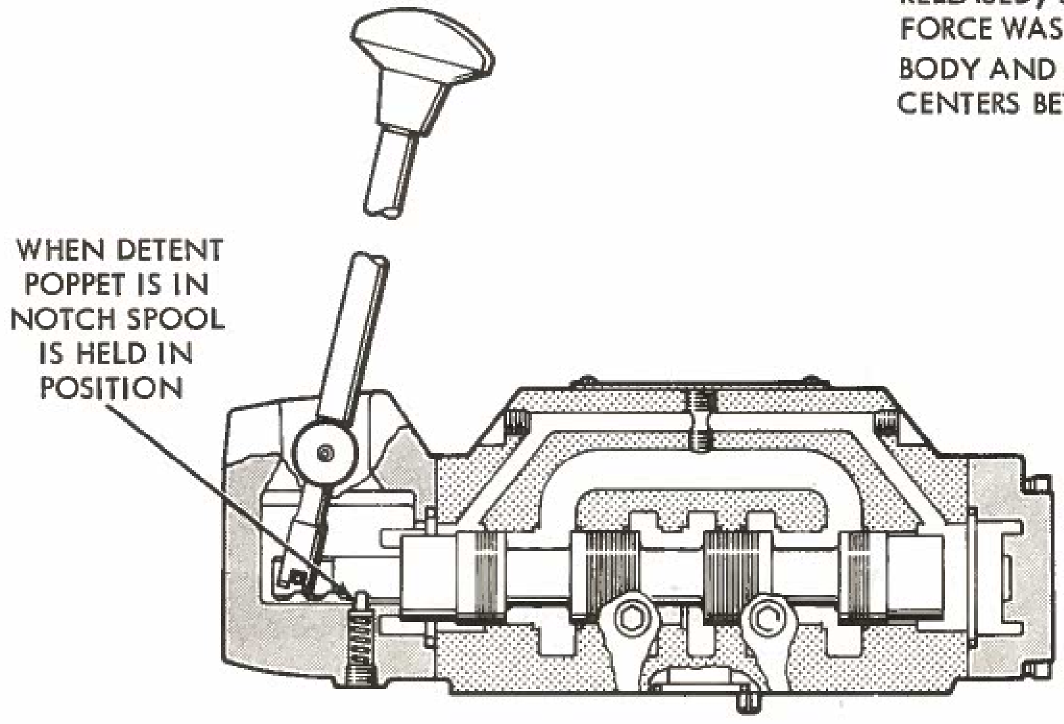

A B
P T
TYPE "0"
T A P B
OPEN CENTER
A B
P T
TYPE "3"
T A P B
PRESSURE AND "B" CLOSED-"A" OPEN TO TANK
A B
P T
TYPE "2"
T A P B
CLOSED CENTER-ALL PORTS CLOSED
A B
P T
TYPE "6"
T A P B
PRESSURE CLOSED-"A" & "B" OPEN TO TANK
A B
P T
TYPE "1"
T A P B
"B" CLOSED-PRESSURE OPEN TO TANK THRU "A"
A B
P T
TYPE "4"
T A P B
TANDEM

Fig. 7-22. Various Center Conditions for Four-Way Valves


1. EXTERNAL CONTROL ACTUATES SPOOL
2. WHEN SPOOL IS ACTUATED, ONE CENTERING SPRING IS COMPRESSED BY WASHER
3. OPPOSITE WASHER IS STATIC AND BUTTS AGAINST VALVE BODY
CENTERING WASHER
4. WHEN CONTROL IS RELEASED, SPRINGS FORCE WASHERS AGAINST BODY AND THE SPOOL CENTERS BETWEEN THEM

SPRING CENTERED



DETENTED

Fig. 7-23. Spool Centering Devices

permitting two or more valves to be connected in series or "tandem."

Spools may be held in their centered positions by centering springs, by spring-loaded detents (Fig. 7-23), or by oil under pressure. The latter provides the fastest and perhaps most positive means of centering.

DG4 SERIES

The DG4 series valves (Fig. 7-24) are sliding spool valves built for direct solenoid operation. Port connections are made by means of a sub-plate permitting easy removal of the valve body for service or replacement. The solenoids are self-contained, push-type units which bolt to the ends of the valve body.

Most of these valves are rated in the 10-20 gpm range. They are built in three-position spring-centered versions and in two position spring-offset or no-spring. Figure 7-25 illustrates the three schematically along with their graphical symbols.

Modifications of this design include a spring-offset, cam-operated version (Fig. 7-26) with an electric limit switch controlled by the spool movement, as well as several different solenoid designs for special applications.

DG3 AND DG5 SERIES VALVES

Larger valves in the DG series are actuated hydraulically, many of them using the DG4 as their pilot. Figure 7-27 illustrates the DG3 pilot operated valve, which also is sub-plate mounted. The spool is shifted by pressure against one end, with the opposite end open to tank. The pressure connections from the remote pilot valve are made through the mounting plate. In some earlier valves, these connections are in the end caps.

DG5 valves are pilot operated, solenoid controlled valves with the pilot valve actually mounted on the main valve body (Fig. 7-28). Both DG3 and DG5 valves are available in spring-centered, no-spring and spring-offset versions (Fig. 7-29) with various spool configurations.

Two-inch and larger valves have flange connections rather than sub-plates and are designated as DF3 (pilot operated) or DF5 (solenoid-controlled, pilot operated).

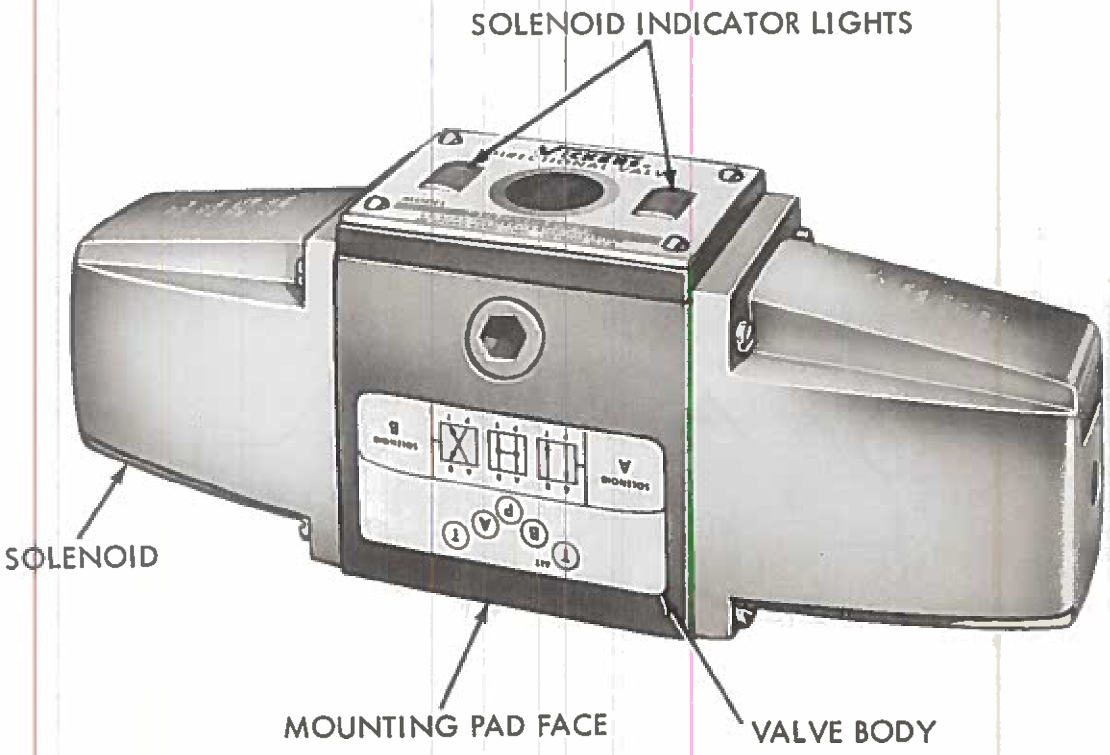
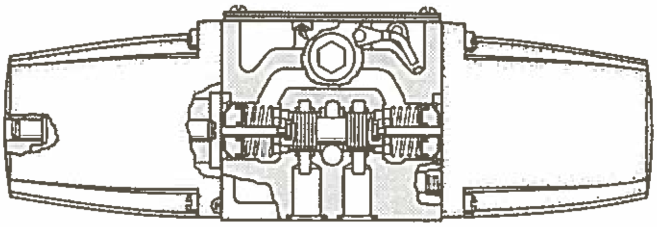
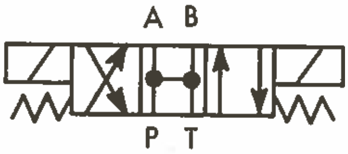


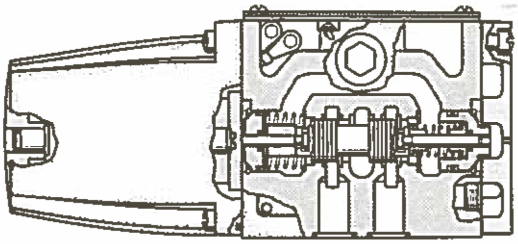
Fig. 7-24. DG4 Series Valves are Solenoid Operated

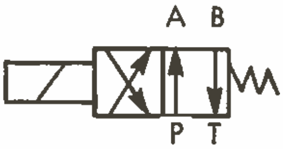
SPRING CENTERED



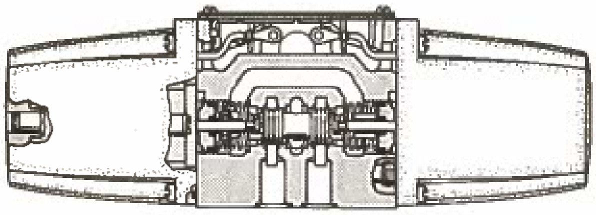


SPRING OFFSET





DETENTED



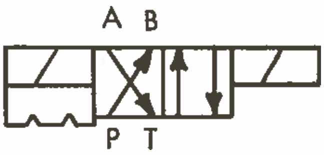


Fig. 7-25. Three Versions of DG Type Four-Way Valve

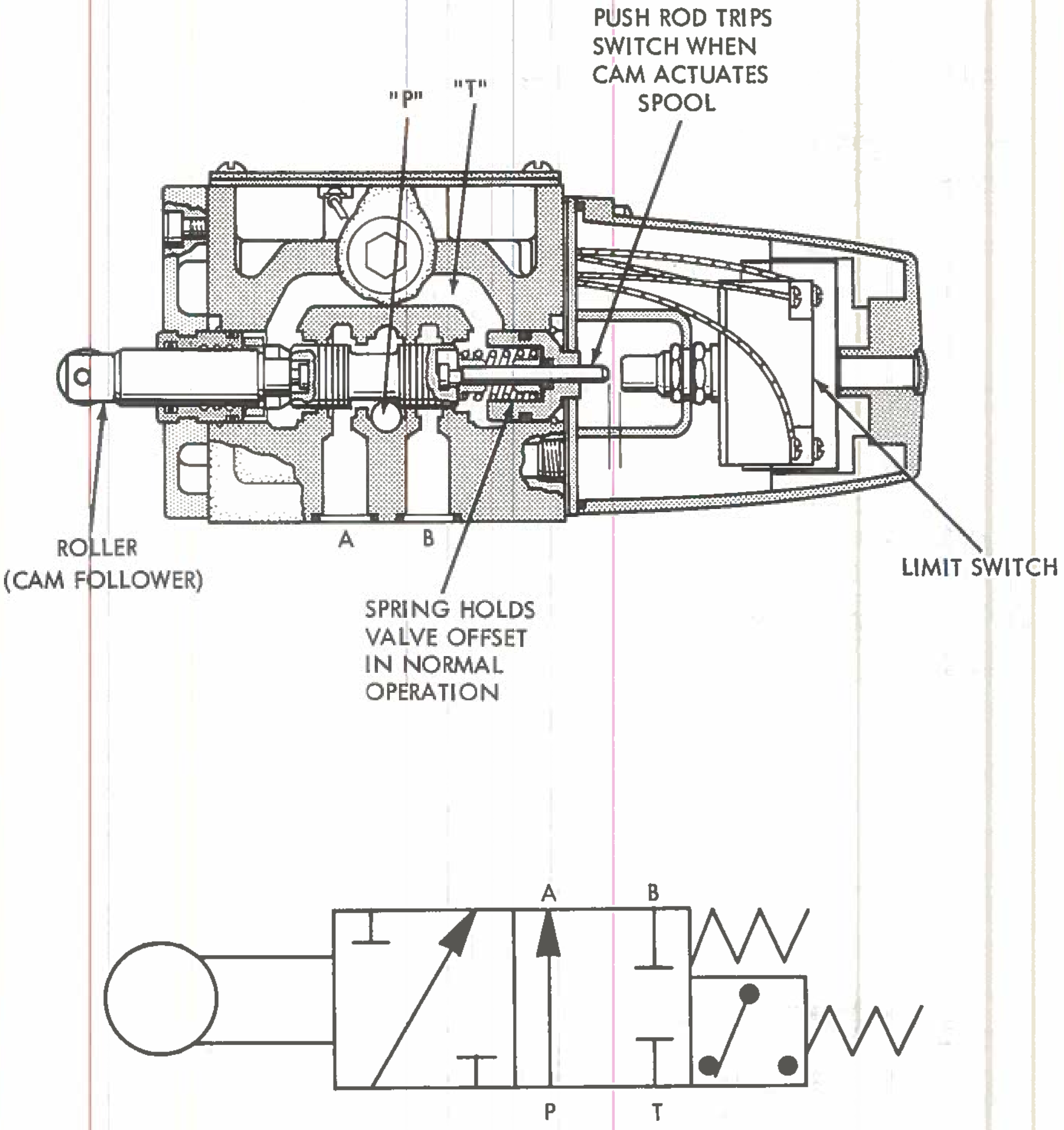

PUSH ROD TRIPS SWITCH WHEN CAM ACTUATES SPOOL
"P"
"T"
ROLLER (CAM FOLLOWER)
A
B
LIMIT SWITCH
SPRING HOLDS VALVE OFFSET IN NORMAL OPERATION
A
B
P
T

Fig. 7-26. Cam-Operated "DG" Valve With Electric Switch

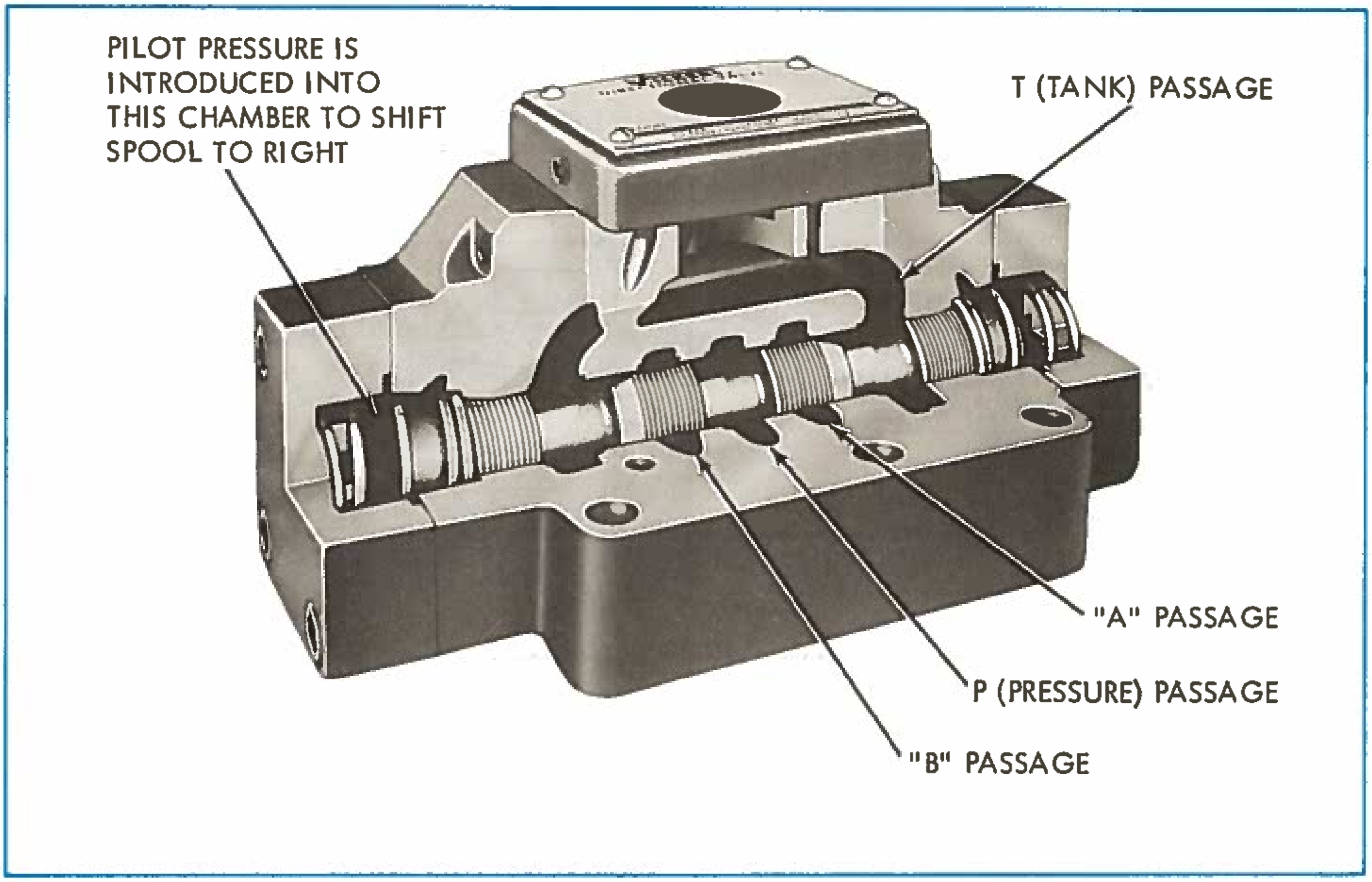

PILOT PRESSURE IS INTRODUCED INTO THIS CHAMBER TO SHIFT SPOOL TO RIGHT
T (TANK) PASSAGE
"A" PASSAGE
P (PRESSURE) PASSAGE
"B" PASSAGE

Fig. 7-27. "DG3" Pilot Operated Valve

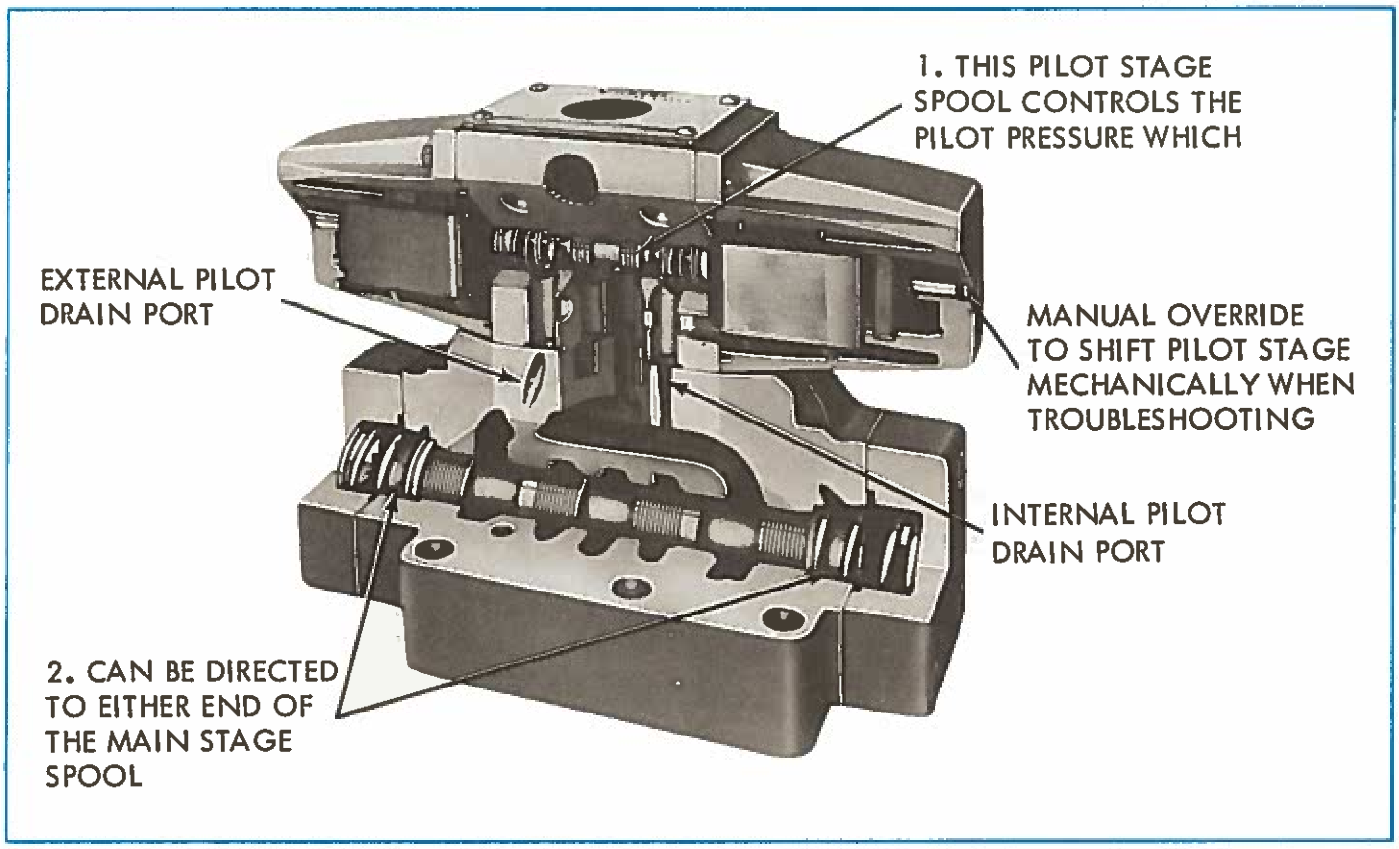

1. THIS PILOT STAGE SPOOL CONTROLS THE PILOT PRESSURE WHICH
EXTERNAL PILOT DRAIN PORT
MANUAL OVERRIDE TO SHIFT PILOT STAGE MECHANICALLY WHEN TROUBLESHOOTING
INTERNAL PILOT DRAIN PORT
2. CAN BE DIRECTED TO EITHER END OF THE MAIN STAGE SPOOL

Fig. 7-28. Typical "DG5" Type Solenoid Controlled, Pilot Operated Valve

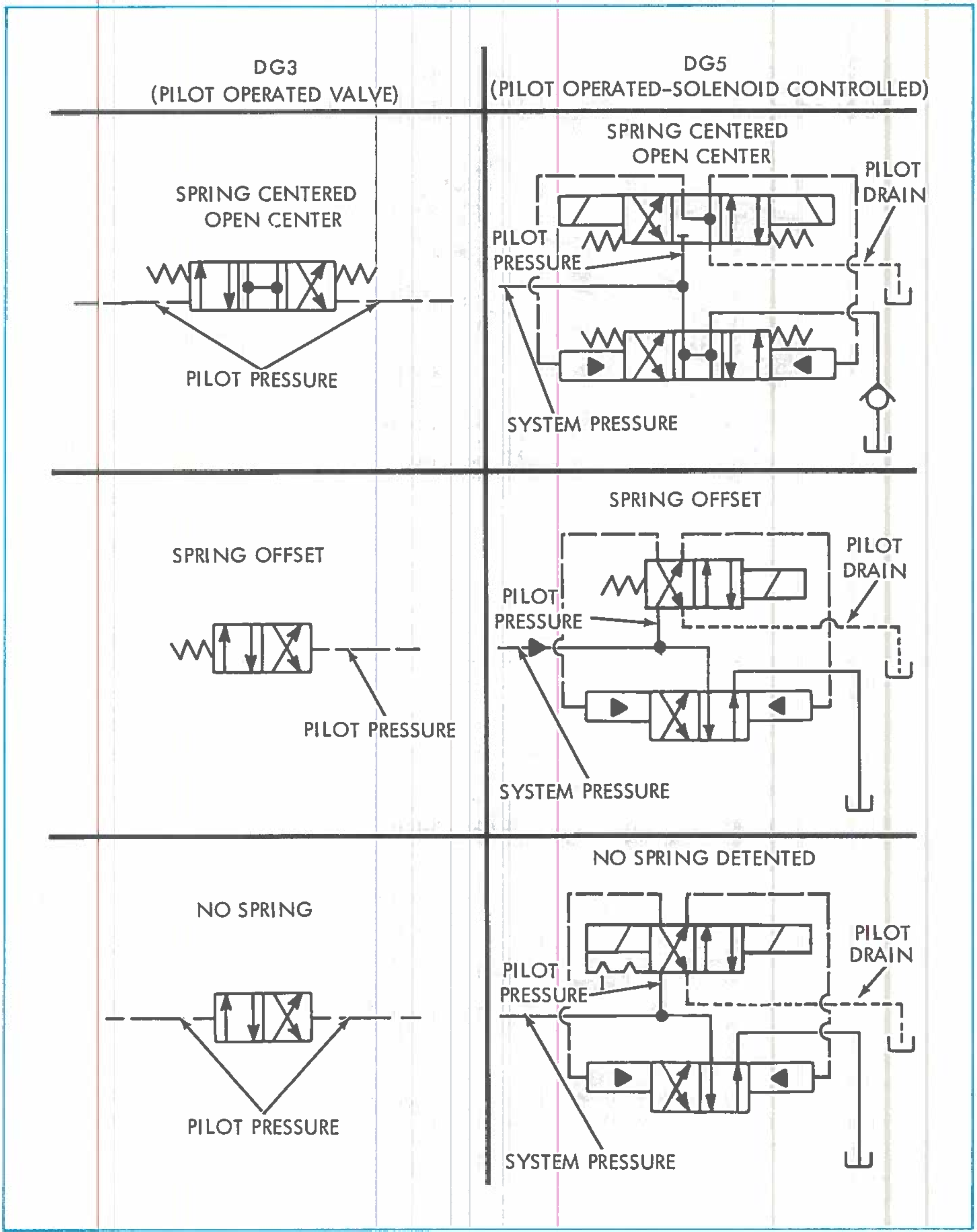


Fig. 7-29. Variations of DG3 and DG5 Valves

PILOT PRESSURE SOURCES

Normally, pilot pressure is supplied in the DG5 valves through an internal passage from the "P" port of the main valve (Fig. 7-29).

Where the pump port is open to tank in the center position, it may be necessary to install a check valve in the tank (return) line to create pilot pressure (Fig. 7-30).

There also are conditions which make it desirable or necessary to use an external source of pilot pressure. In this case, the internal pilot connection is plugged, and pilot oil is supplied through a separate port in the sub-plate. A connection ahead of a check valve installed upstream from the directional valve as shown in Figure 7-30 is one method of accomplishing this. However, some models of the DG5 valve are available with a check valve installed in the pressure port of the body for this purpose. The pilot oil is then available internally. See Figure 7-31.

PILOT CHOKE

A pilot choke (Fig. 7-32) may be incorporated to slow the spool travel for smoother reversals or to provide a brief time delay or dwell period before the actuator is reversed.

The pilot choke functions in effect as a meter-out restriction valve. It allows free pilot flow to the end of the main spool, but restricts flow out of opposite end; thus reversing flow gradually and cushioning the spool's contact when it shifts. The controlling orifices are adjustable. Free flow in is accomplished by check valves.

A pilot choke assembly is available for mounting directly on a DG3 or DF3 valve, or between the pilot and main valves in a DG5 or DF5 valve (Fig. 7-33). In other valves the choke may be built into the end caps.

PILOT PISTONS

Pilot pistons (Fig. 7-34) are sometimes used when large valve spools are shifted hydraulically. Since it is only necessary to fill the volume displaced by the small piston, less pilot oil is required and faster shift times can be attained.

A differential piston is simply the incorporation of a single piston on one end to provide differential areas. Constant pressure applied to the smaller area of the differential piston may be used to bias the spool to one side in place of a spring.

DECELERATION VALVES

Hydraulic cylinders often have cushions built in to slow down the cylinder pistons at the extreme ends of their travel. When it is necessary to decelerate a cylinder at some intermediate position or to slow down or stop a rotary actuator (motor), an external valve is required.

Most deceleration valves are cam operated valves with tapered spools. They are used to gradually decrease flow to or from an actuator for smooth stopping or deceleration. A "normally open" valve cuts off flow when its plunger is depressed by a cam. It may be used to slow a drill head cylinder down at the transition from rapid traverse to feed or to stop heavy index tables and large presses smoothly.

Some applications require a valve to permit flow when it is actuated and to cut off flow when the plunger is released. In this case a "normally closed" valve is used. This type valve often is used to provide an interlocking arrangement whereby flow can be directed to another branch of the circuit when the actuator or load reaches a certain position. Both the "normally open" and "normally closed" type valves are available with integral check valves to permit reverse free flow.

TAPERED PLUNGER DESIGN

An early design of deceleration valve (Fig. 7-35) uses a tapered plunger to reduce flow as it is actuated by the cam. Before the plunger is depressed (view A), free flow is permitted from the inlet to the outlet. Depressing the plunger gradually cuts the flow off (view B). Reverse free flow (view C) is permitted by the integral check valve.

The control range of this valve depends on the volume of flow and on the cam rise. At nearly maximum volume; that is, with an initial pressure drop through the valve, there is control throughout the plunger stroke. At low flow rates, only part of the travel is available for control--from the point where a pressure drop is created.

This drawback has been overcome in the adjustable orifice design valve which permits tailoring of the valve to any given flow.

ADJUSTABLE ORIFICE DESIGN

The adjustable-orifice design valve, model series DT15S2, is illustrated in Figure 7-36. In this valve, a closely-fitted plunger and sleeve with rectangular ports or windows are used to control flow.

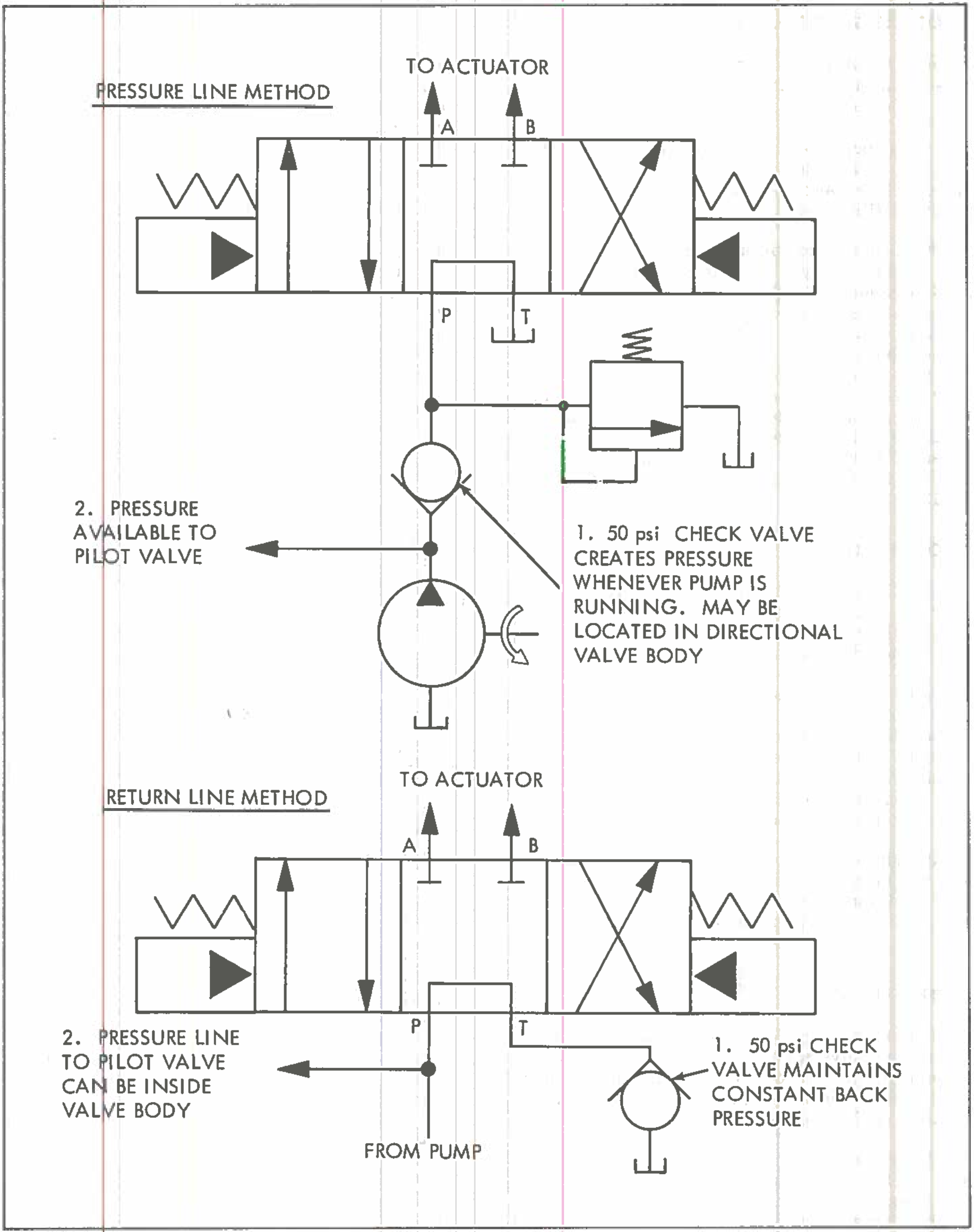

PRESSURE LINE METHOD
TO ACTUATOR
A
B
P
T
2. PRESSURE AVAILABLE TO PILOT VALVE
1. 50 psi CHECK VALVE CREATES PRESSURE WHENEVER PUMP IS RUNNING. MAY BE LOCATED IN DIRECTIONAL VALVE BODY
RETURN LINE METHOD
TO ACTUATOR
A
B
P
T
2. PRESSURE LINE TO PILOT VALVE CAN BE INSIDE VALVE BODY
1. 50 psi CHECK VALVE MAINTAINS CONSTANT BACK PRESSURE
FROM PUMP

Fig. 7-30. Pilot Pressure Check Valve

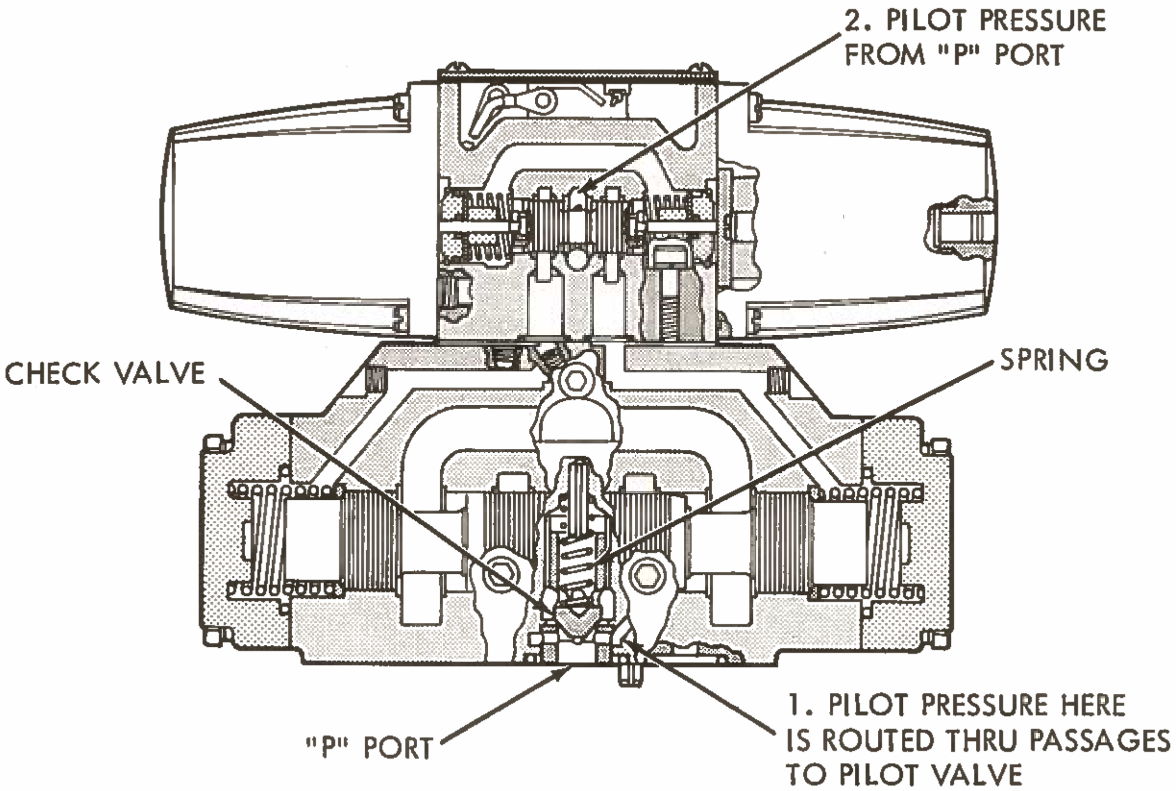

2. PILOT PRESSURE FROM "P" PORT
CHECK VALVE
SPRING
"P" PORT
1. PILOT PRESSURE HERE IS ROUTED THRU PASSAGES TO PILOT VALVE

Fig. 7-31 -- Integral Pilot Pressure Check Valve

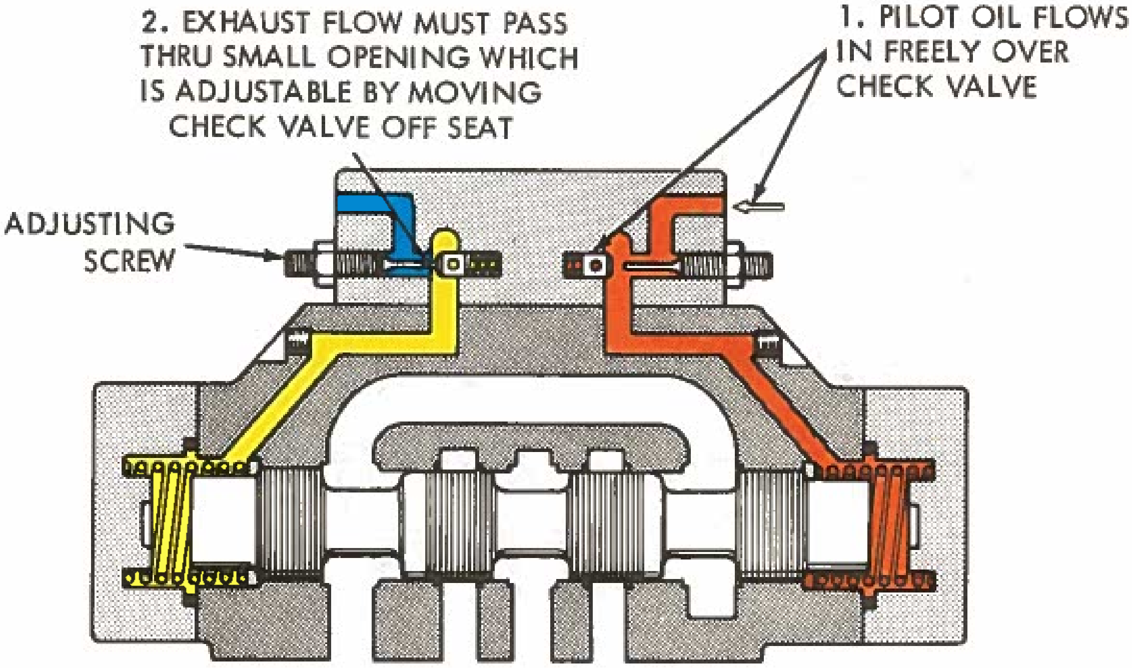

2. EXHAUST FLOW MUST PASS THRU SMALL OPENING WHICH IS ADJUSTABLE BY MOVING CHECK VALVE OFF SEAT
1. PILOT OIL FLOWS IN FREELY OVER CHECK VALVE
ADJUSTING SCREW

Fig. 7-32. Pilot Choke Controls Spool Movement

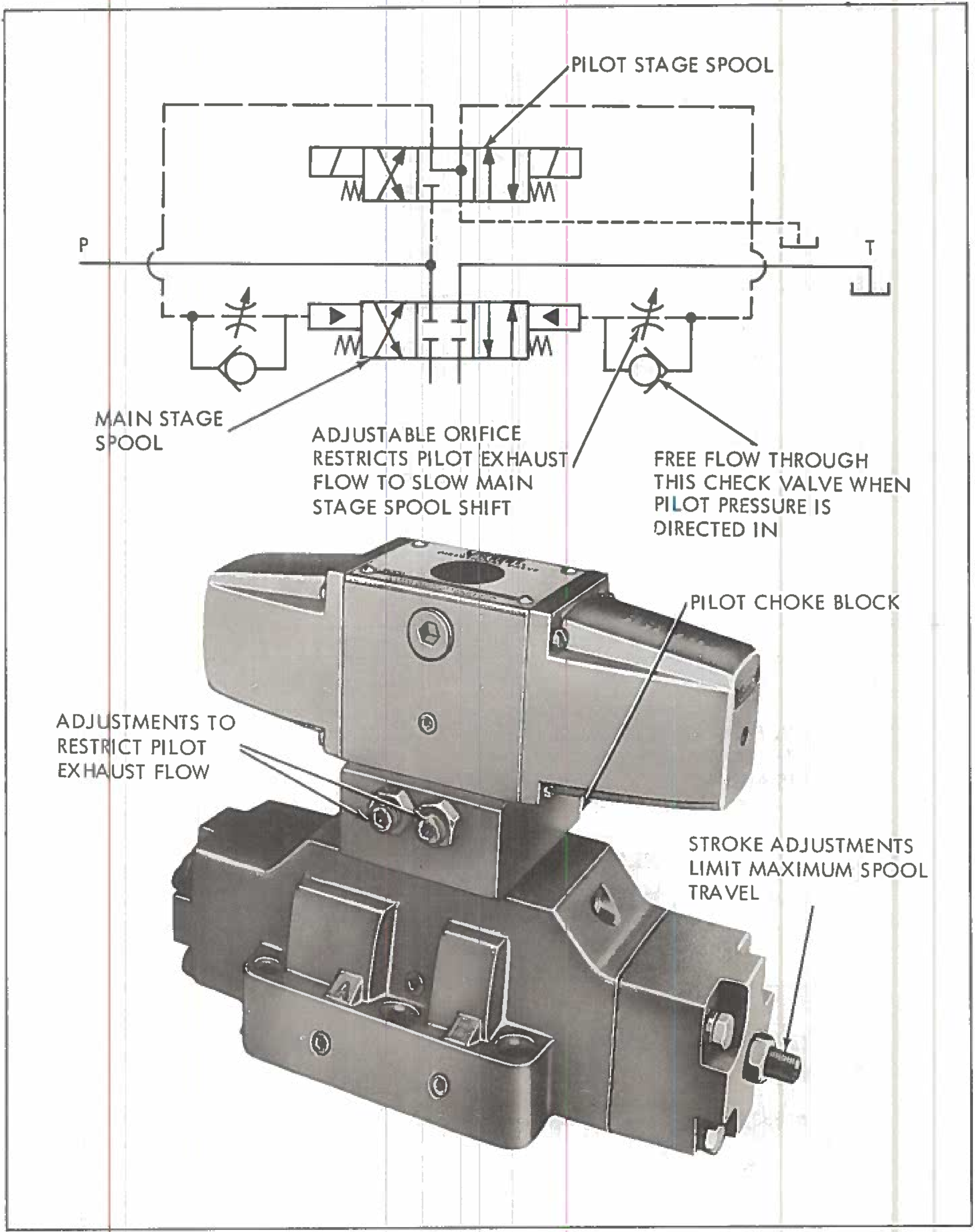

PILOT STAGE SPOOL
P
T
MAIN STAGE SPOOL
ADJUSTABLE ORIFICE RESTRICTS PILOT EXHAUST FLOW TO SLOW MAIN STAGE SPOOL SHIFT
FREE FLOW THROUGH THIS CHECK VALVE WHEN PILOT PRESSURE IS DIRECTED IN
PILOT CHOKE BLOCK
ADJUSTMENTS TO RESTRICT PILOT EXHAUST FLOW
STROKE ADJUSTMENTS LIMIT MAXIMUM SPOOL TRAVEL

Fig. 7-33. Pilot Choke Mounts on DG3 or DG5 Valve

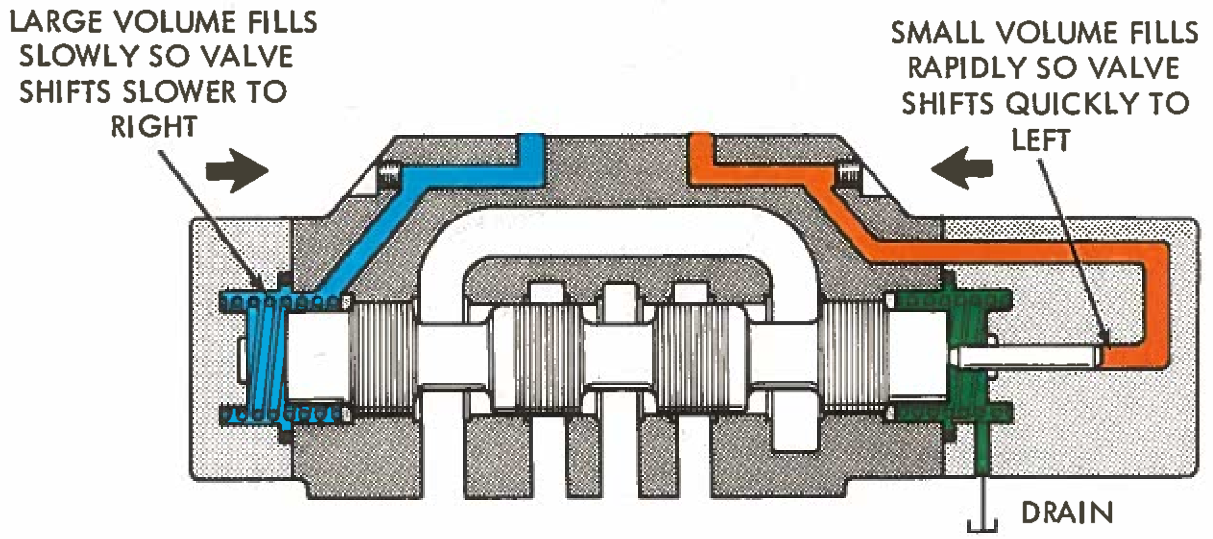

LARGE VOLUME FILLS SLOWLY SO VALVE SHIFTS SLOWER TO RIGHT
SMALL VOLUME FILLS RAPIDLY SO VALVE SHIFTS QUICKLY TO LEFT
DRAIN

Fig. 7-34. Pilot Piston Speeds Valve Shifting

The plunger moves inside the sleeve, and the ports in each coincide in the open position. Oil entering the inlet flows through the small upper ports in the sleeve and plunger, down through the center of the plunger and out the large ports to the outlet (view A). When the plunger is depressed, the "window" area is gradually cut off to stop flow (view B). Reverse free flow is allowed by the integral check valve.

Initial Pressure Drop Adjustment

For precise control throughout the plunger stroke, the width of the "window" openings is controlled by adjusting screws which turn the sleeve. For low flow rates, the opening is narrow; for higher flow rates, it is wide. The adjustment is made by attaching a pressure gauge at the side of the valve and turning the screws to obtain the desired initial pressure drop.

This valve also includes an adjustable orifice which allows some flow with the plunger fully depressed. This permits the load to creep to its final position in indexing and similar applications. It consists of a small plunger with a chamfered end and "vee" notch which can be set to by-pass the spool-sleeve closure.

The window orifice valve is built in both pipe-threaded and back-mounted versions. Both valves require a drain to permit leakage oil to escape from beneath the plunger.

TYPICAL APPLICATIONS

Figure 7-37 illustrates a typical application of a deceleration valve. Here it is used to slow a drill head cylinder from rapid advance speed to feed speed at a preset point. View A shows rapid advance, with exhaust flow from the cylinder passing through the deceleration valve unrestricted.

In view B, we see the valve plunger being depressed by a cam. Exhaust flow is cut off at the deceleration valve and must pass through the volume control valve, which sets the feed speed. In view C, the directional valve has been reversed to return the cylinder. Oil from the directional valve passes through the deceleration valve freely over the check valve, whether or not the plunger is depressed.

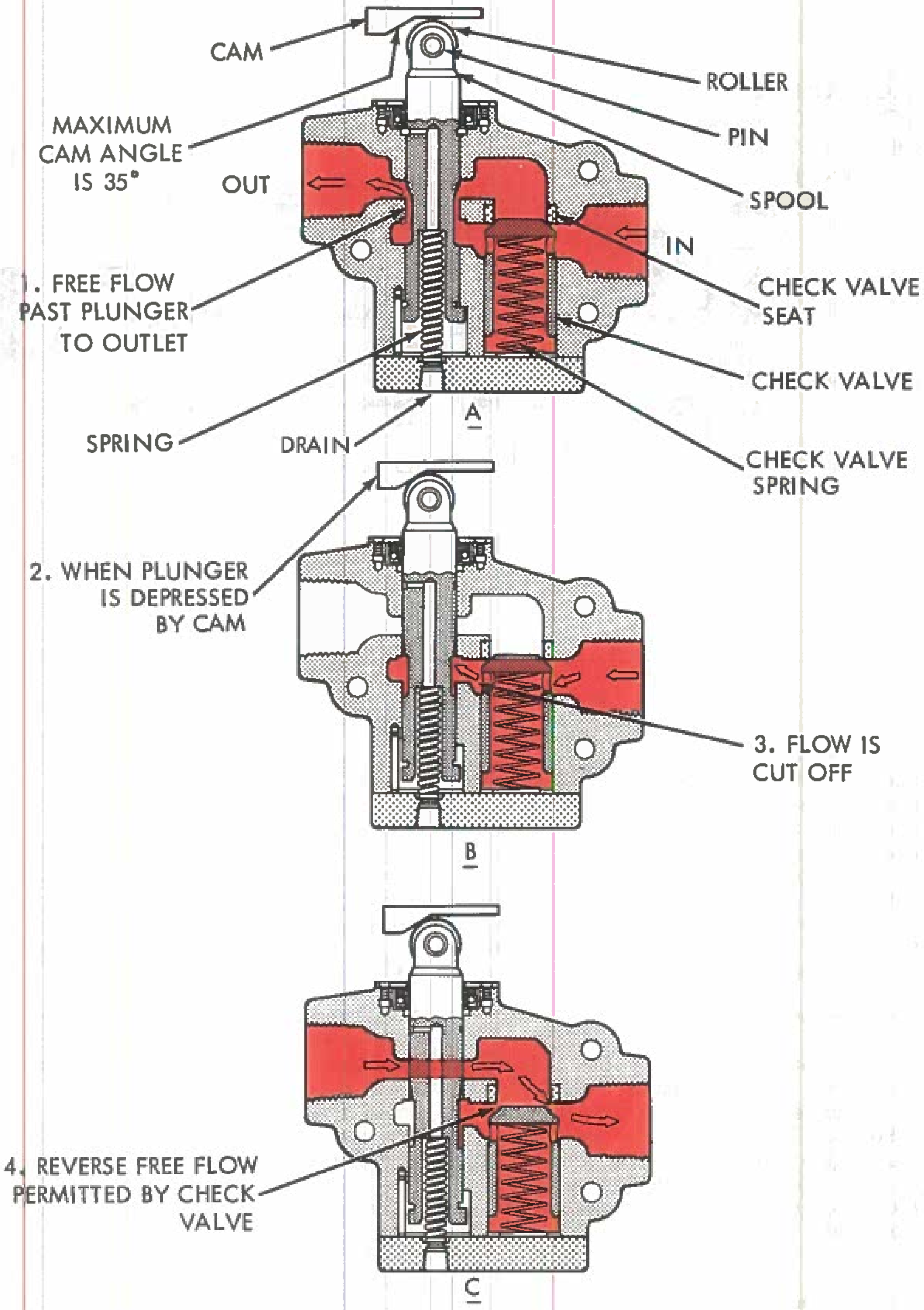


Fig. 7-35. C-700 Series Deceleration Valve Has Tapered Plunger

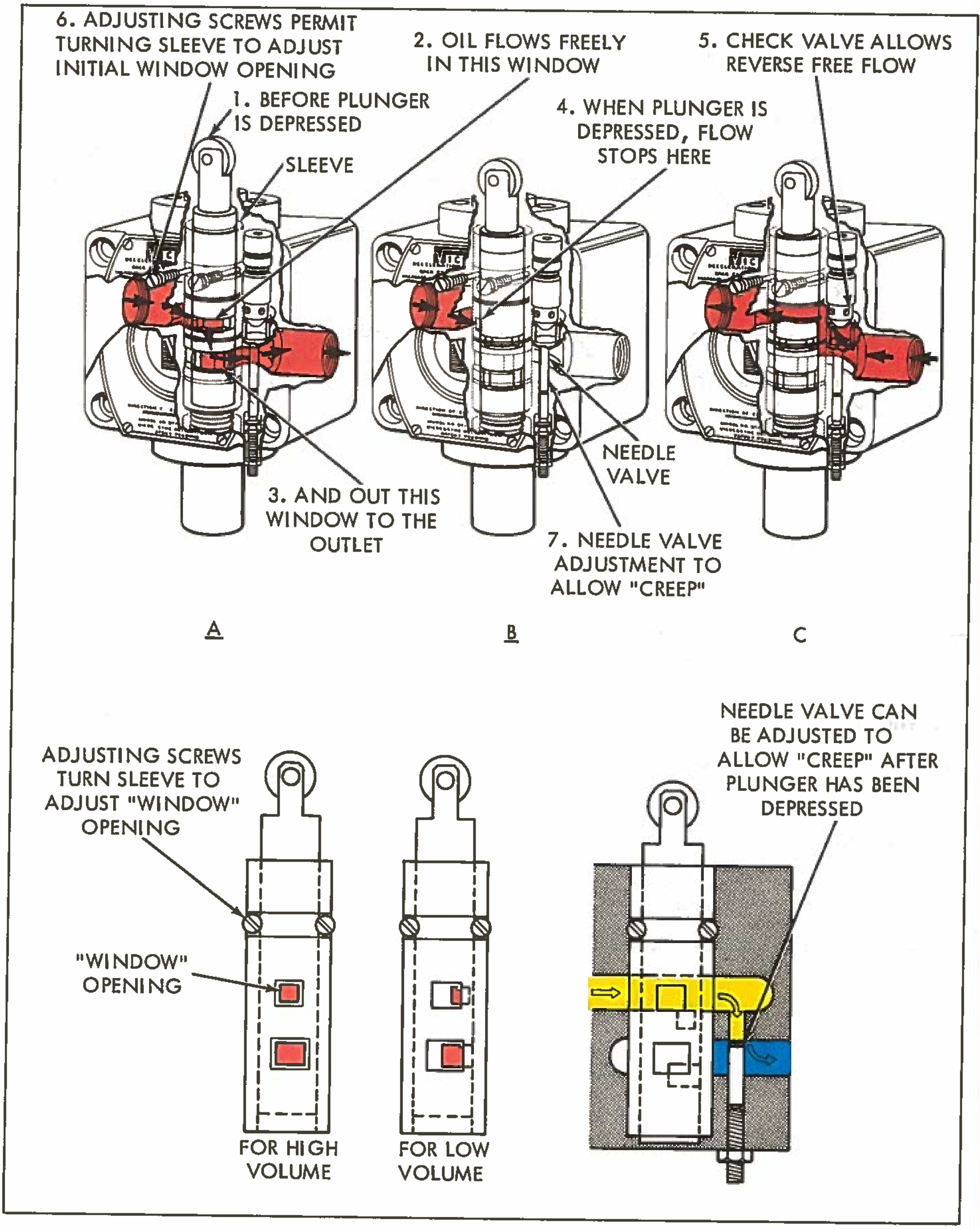

6. ADJUSTING SCREWS PERMIT TURNING SLEEVE TO ADJUST INITIAL WINDOW OPENING
2. OIL FLOWS FREELY IN THIS WINDOW
5. CHECK VALVE ALLOWS REVERSE FREE FLOW
1. BEFORE PLUNGER IS DEPRESSED
4. WHEN PLUNGER IS DEPRESSED, FLOW STOPS HERE
SLEEVE
NEEDLE VALVE
3. AND OUT THIS WINDOW TO THE OUTLET
7. NEEDLE VALVE ADJUSTMENT TO ALLOW "CREEP"
A
B
C
NEEDLE VALVE CAN BE ADJUSTED TO ALLOW "CREEP" AFTER PLUNGER HAS BEEN DEPRESSED
ADJUSTING SCREWS TURN SLEEVE TO ADJUST "WINDOW" OPENING
"WINDOW" OPENING
FOR HIGH VOLUME
FOR LOW VOLUME

Fig. 7-36. This Deceleration Valve Has a "Window" Orifice In Place of Plunger Taper

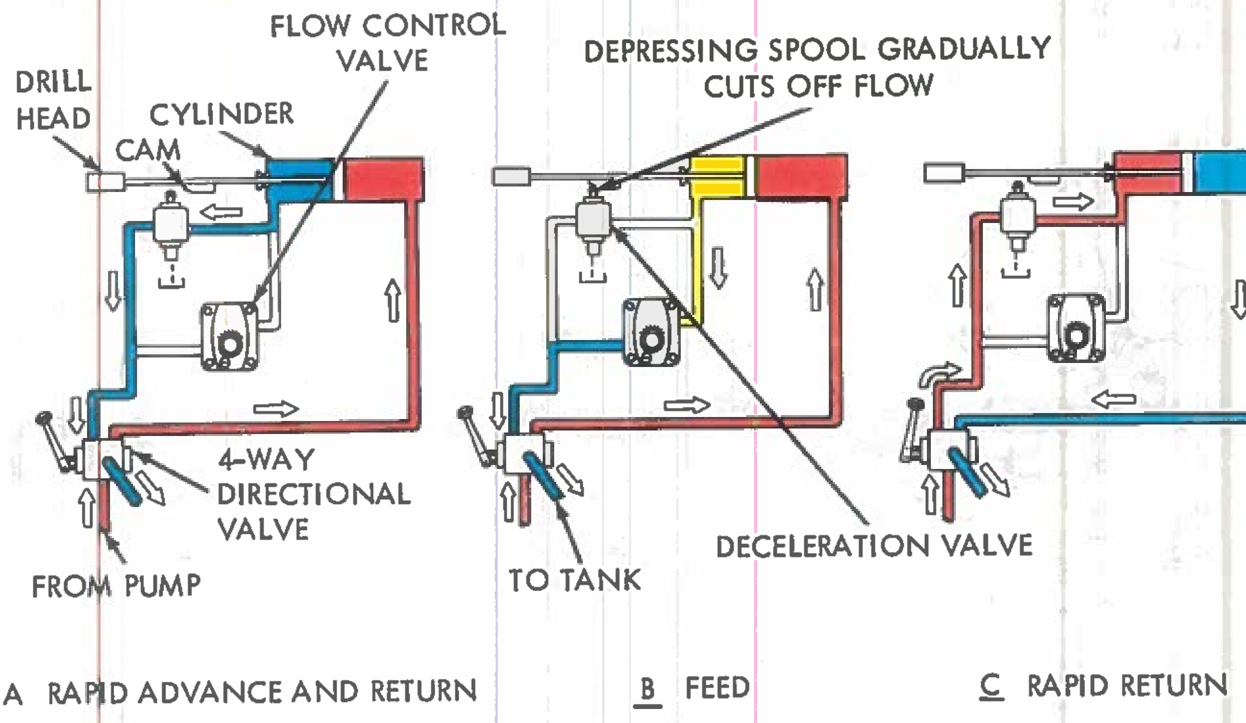


Fig. 7-37. Deceleration Valve In Feed Control System

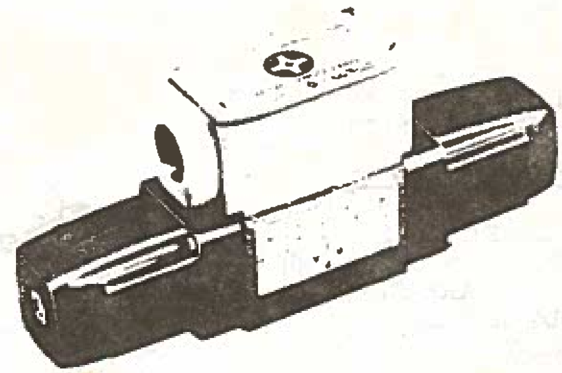
QUESTIONS

1. What is the function of a directional control?
2. What is meant by finite positioning?
3. Explain the function of a check valve.
4. What is the operational difference between "2C" and "4C" pilot-operated check valves?
5. What type of directional valve is used to reverse an actuator?
6. Name three ways to shift a four-way valve.
7. How many positions does a spring-offset valve have? A spring-centered valve?
8. Describe the "centered" condition of a tandem four-way valve.
9. How may pilot pressure be created for an open-center, pilot-operated valve?
10. What is the function of a pilot choke?
11. How is the adjustable-orifice deceleration valve an improvement over the tapered-plunger design?

VICKERS

②

SOLENOID OPERATED DIRECTIONAL VALVE



this complete unit can be replaced at
nominal cost, factory repair is not
practical. Kits are available from our
Branch Warehouse system to support
customer repair of this unit.

SERVICE PARTS INFORMATION

DG4V-3-* A -*(*)-*-10 OFFSET
DG4V-3-* B -*(*)-*-10 SPRING CENTERED
DG4V-3-* C -*(*)-*-10 SPRING CENTERED
DG4V-3-* N -*(*)-*-11 DETENTED

SPERRY VICKERS, TROY, MI 48084
VICKERS, INCORPORATED, IS A SUBSIDIARY OF SPERRY CORPORATION
SPERRY VICKERS IS A REGISTERED TRADEMARK OF SPERRY CORPORATION

Revised 9-1-83

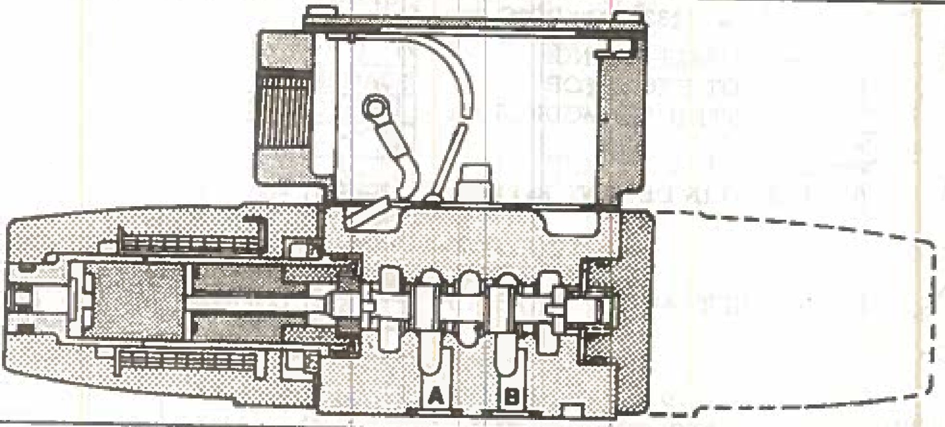
I-3529-S

MODEL CODE BREAKDOWN

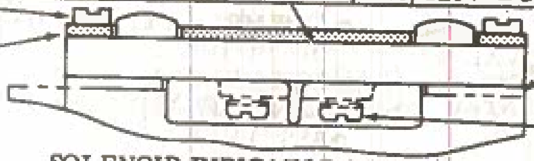
DG4V-3--W(L)-*-10-(LH)-(S300)**

- DIRECTIONAL VALVE
- SUBPLATE OR MANIFOLD MOUNTED
- SOLENOID OPERATED
- MAXIMUM PRESSURE RATING 350 bar RATING (5000 PSI)
- NFPA-D01 INTERFACE
- SPOOL TYPE
- A - SPRING OFFSET
  B - SPRING CENTERED SINGLE SOLENOID
  C - SPRING CENTERED
  N - NO SPRING DETENTED
- OPTIONS
  W-½ NPT WIRING HOUSING
  WL-½ NPT WIRING HOUSING WITH SOLENOID INDICATOR LIGHTS
- COIL CODE LETTER (SEE TABLE FOR VOLTAGE & FREQUENCY)
- DESIGN
  -11 DESIGN "N" MODELS
- SINGLE SOLENOID MODELS ONLY. FOR LEFT HAND ASSEMBLY. OMIT FOR RIGHT HAND ASSEMBLY
- 210 bar, (3045 PSI) OPERATION ALL PORTS)

NOTE: (F3) VITON SEALS ARE STANDARD IN THIS UNIT.



SOLENOID INDICATOR LIGHTS		
△INCLUDED IN SERVICE KIT	△LIGHT S/A	VOLTAGE RANGE
942385	751255	100/127V-AC
942386	751256	192/233V-AC
942490	751881	110/120V-DC



□△751257 SCREW (4 REQ'D)

△NAMEPLATE (STAMP MODEL CODE ON NAMEPLATE BEFORE ASSEMBLY)

△422003 WIRE & GASKET S/A

417358 SCREWS (4 REQ'D) (REFERENCE ONLY)

SOLENOID INDICATOR LIGHT(S)

To insure sustained efficiency and maximum trouble-free life of this precision equipment, initial and continuous filtration of the fluid medium to 35 microns absolute or less is essential. (For information pertaining to Sperry Vickers economical filters, see bulletin 81-216.)

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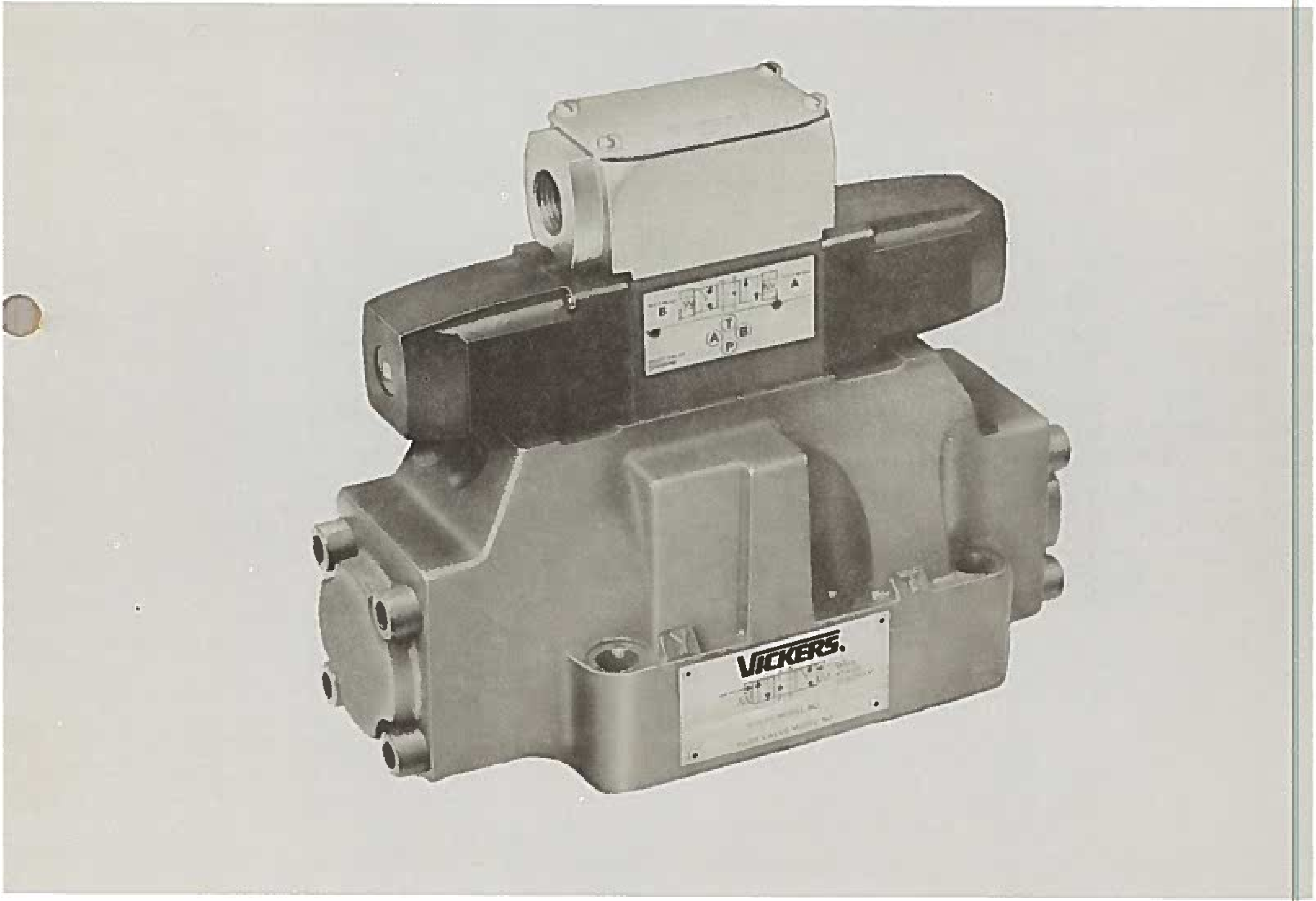
VICKERS

3

Service Parts Information

Pilot Operated Solenoid Controlled Directional Valve

DG5S-8-*A-***-10/11
DG5S-8-*B-***-10/11
DG5S-8-*C-***-10/11
DG5S-8-*N-***-10/11



Vickers, Incorporated

1401 Crooks Road
Troy, Michigan 48084

REVISED 12-1-84

I-3439-S

PLUG INSTALLATION TABLE			
MODEL	"A"PLUG	"B"PLUG	"C"PLUG
DG5S-8-** -1*	DOES NOT EXIST	237588	———
DG5S-8-**-E -1*		113000	237588
DG5S-8-**X -1*		———	———
DG5S-8-**X -E -1*		113000	
DG5S-8-**-KRS-1*	237588	DOES NOT EXIST	237588
DG5S-8-**-E-KRS-1*	113000		
DG5S-8-**X-KRS-1*	———		———
DG5S-8-**X-E-KRS-1*	113000		
∗113000 SOLID PLUG	∗237588 ORIFICE PLUG		

VALVE MODEL CODE		SPOOL	MAIN STAGE ID PLATE W/ CIRCUIT DIAGRAM	
MAIN STAGE SPOOL	AVAILABLE IN VALVE TYPE		"A"ONLY	B/C/N
DG5S-8-0	A/B/C/N	363495	400975	400976
DG5S-8-1	B/C	276623*	———	400977
DG5S-8-2	A/B/C/N	363496	400975	400978
DG5S-8-3	B/C	276625*	———	400979
DG5S-8-4	B/C	276626		400980
DG5S-8-6	A/B/C/N	363498	400975	400981
DG5S-8-7	B/C	345589	———	573685
DG5S-8-8	B/C	363499	———	400980
DG5S-8-9	A/B/C/N	363500	400975	400976
DG5S-8-11	B/C	276623*		632700
DG5S-8-31	B/C	276625*	———	580475
DG5S-8-33	A/B/C/N	363501	400975	400981

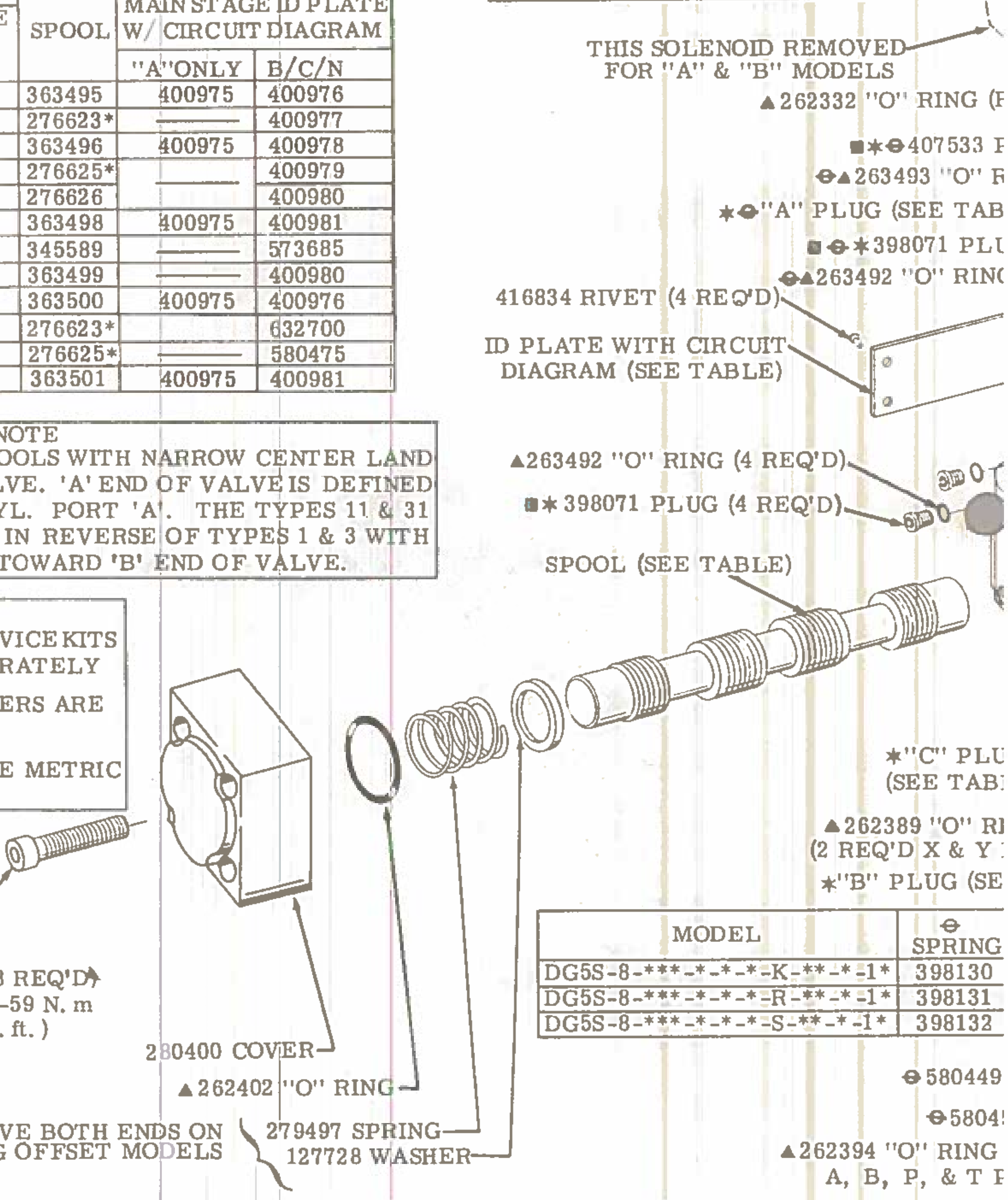
* NOTE

ASSEMBLE TYPE 1 & 3 SPOOLS WITH NARROW CENTER LAND TOWARDS 'A' END OF VALVE. 'A' END OF VALVE IS DEFINED AS BEING CLOSEST TO CYL. PORT 'A'. THE TYPES 11 & 31 SPOOLS ARE INSTALLED IN REVERSE OF TYPES 1 & 3 WITH NARROW CENTER LANDS TOWARD 'B' END OF VALVE.

NOTE

- PARTS INCLUDED IN SERVICE KITS WILL NOT BE SOLD SEPARATELY
- ALL THREADED FASTENERS ARE METRIC
- END COVER SCREWS ARE METRIC GRADE 12. 9

PILOT STAGE ATTACHING BOLTS	
MODEL	BOLT KIT
W/OUT PILOT CHOKE	255699
W/PILOT CHOKE	466838
BOLT KIT INCLUDES 4 BOLTS	
TORQUE TO 4. 5 - 5. 7 N. m (39. 8 - 50. 4 lb. in.)	
. SEE BACK PAGE FOR PILOT CHOKE PARTS BREAKDOWN	

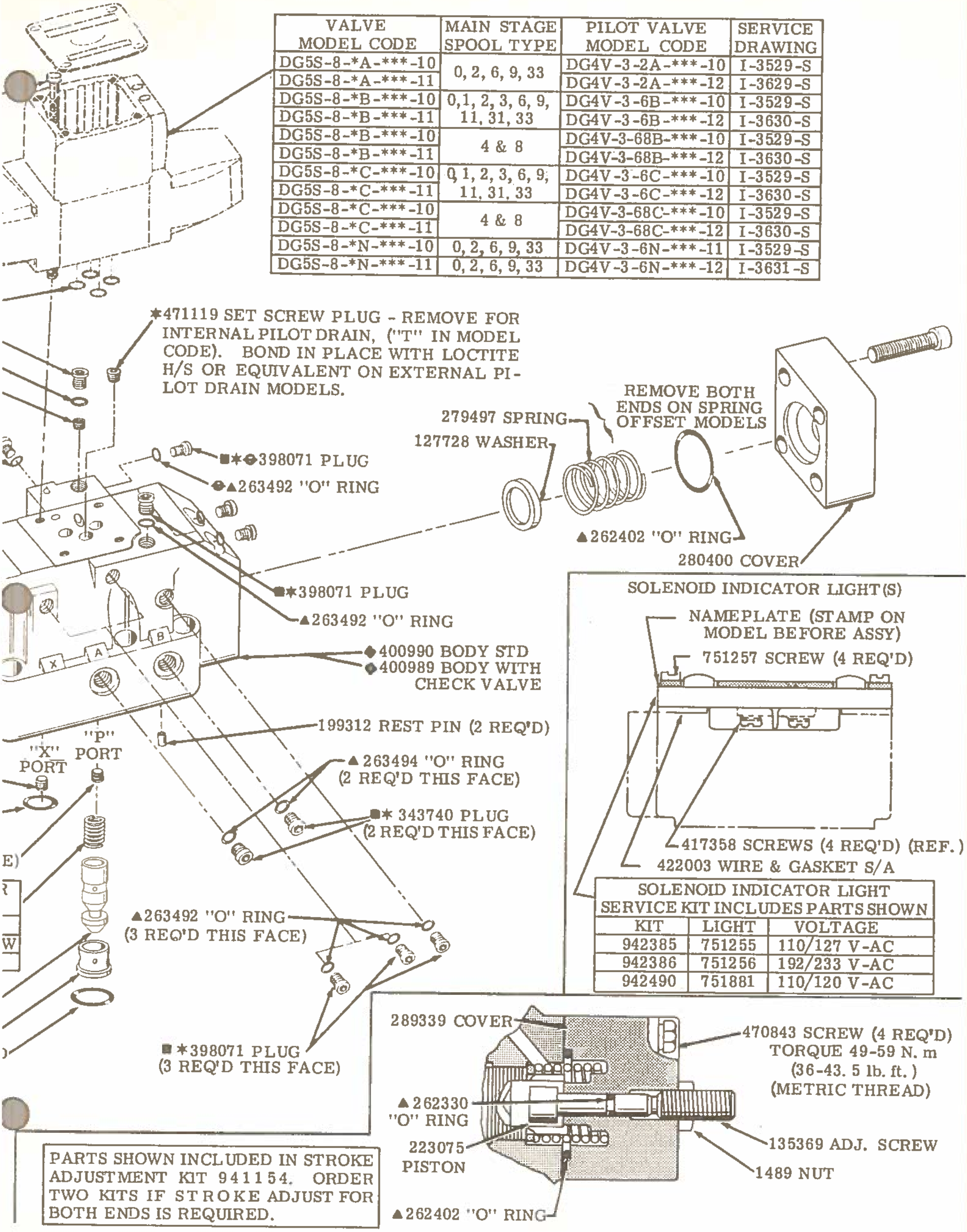


MODEL	⊖ SPRING
DG5S-8-***-*-*-*-K-**-*-1*	398130
DG5S-8-***-*-*-*-R-**-*-1*	398131
DG5S-8-***-*-*-*-S-**-*-1*	398132

▲INCLUDED IN F3 SEAL KIT 920148
∗INCLUDED IN PLUG KIT 941167
◇INCLUDED IN FASTENER KIT 941165
◆NOT AVAILABLE FOR SALE
⊖USED ON CHECK VALVE MODELS ONLY
■PLUG TORQUES (SEE TABLE)

PLUG TORQUES (OILED)		
PLUG	TORQUE (N. m)	TORQUE (lb. in.)
113000	5. 0-5. 9	45-52
343740	15. 0-16. 0	133-143
398071	9. 8-10. 2	67-90
407533	12. 1-12. 4	107-109

VALVE MODEL CODE	MAIN STAGE SPOOL TYPE	PILOT VALVE MODEL CODE	SERVICE DRAWING
DG5S-8-*A-***-10	0, 2, 6, 9, 33	DG4V-3-2A-***-10	I-3529-S
DG5S-8-*A-***-11		DG4V-3-2A-***-12	I-3629-S
DG5S-8-*B-***-10	0,1, 2, 3, 6, 9, 11, 31, 33	DG4V-3-6B-***-10	I-3529-S
DG5S-8-*B-***-11		DG4V-3-6B-***-12	I-3630-S
DG5S-8-*B-***-10	4 & 8	DG4V-3-68B-***-10	I-3529-S
DG5S-8-*B-***-11		DG4V-3-68B-***-12	I-3630-S
DG5S-8-*C-***-10	0, 1, 2, 3, 6, 9, 11, 31, 33	DG4V-3-6C-***-10	I-3529-S
DG5S-8-*C-***-11		DG4V-3-6C-***-12	I-3630-S
DG5S-8-*C-***-10	4 & 8	DG4V-3-68C-***-10	I-3529-S
DG5S-8-*C-***-11		DG4V-3-68C-***-12	I-3630-S
DG5S-8-*N-***-10	0, 2, 6, 9, 33	DG4V-3-6N-***-11	I-3529-S
DG5S-8-*N-***-11	0, 2, 6, 9, 33	DG4V-3-6N-***-12	I-3631-S


✱471119 SET SCREW PLUG - REMOVE FOR INTERNAL PILOT DRAIN, ("T" IN MODEL CODE). BOND IN PLACE WITH LOCTITE H/S OR EQUIVALENT ON EXTERNAL PILOT DRAIN MODELS.
REMOVE BOTH ENDS ON SPRING OFFSET MODELS
279497 SPRING
127728 WASHER
■✱⊖398071 PLUG
⊖▲263492 "O" RING
▲262402 "O" RING
280400 COVER
■✱398071 PLUG
▲263492 "O" RING
◆400990 BODY STD
◈400989 BODY WITH CHECK VALVE
"X" PORT
"P" PORT
X
A
B
199312 REST PIN (2 REQ'D)
▲263494 "O" RING (2 REQ'D THIS FACE)
■✱ 343740 PLUG (2 REQ'D THIS FACE)
▲263492 "O" RING (3 REQ'D THIS FACE)
■✱398071 PLUG (3 REQ'D THIS FACE)
SOLENOID INDICATOR LIGHT(S)
NAMEPLATE (STAMP ON MODEL BEFORE ASSY)
751257 SCREW (4 REQ'D)
417358 SCREWS (4 REQ'D) (REF.)
422003 WIRE & GASKET S/A
289339 COVER
470843 SCREW (4 REQ'D) TORQUE 49-59 N. m (36-43. 5 lb. ft.) (METRIC THREAD)
▲262330 "O" RING
223075 PISTON
135369 ADJ. SCREW
1489 NUT
▲262402 "O" RING

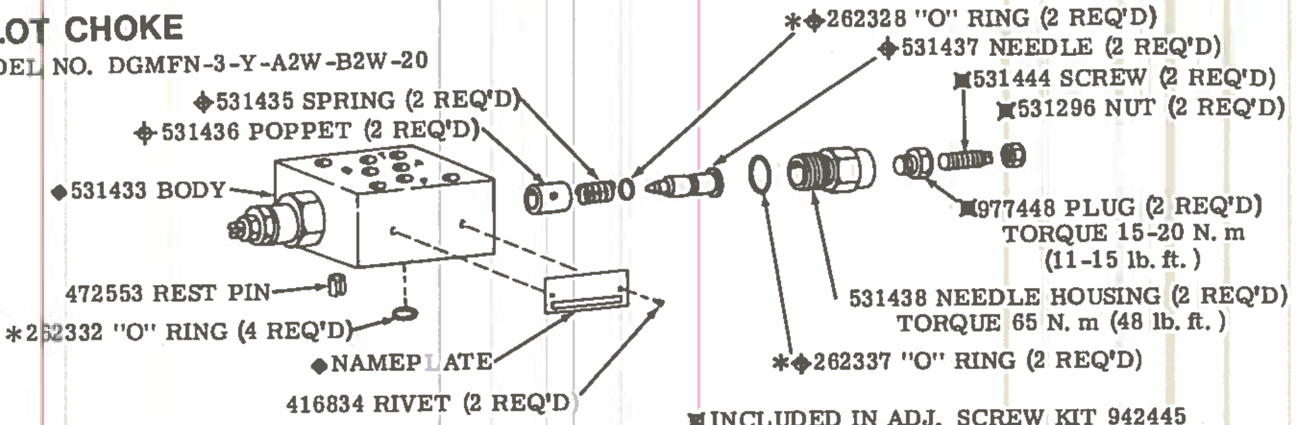
SOLENOID INDICATOR LIGHT SERVICE KIT INCLUDES PARTS SHOWN

KIT	LIGHT	VOLTAGE
942385	751255	110/127 V-AC
942386	751256	192/233 V-AC
942490	751881	110/120 V-AC

PARTS SHOWN INCLUDED IN STROKE ADJUSTMENT KIT 941154. ORDER TWO KITS IF STROKE ADJUST FOR BOTH ENDS IS REQUIRED.
)

PILOT CHOKE

MODEL NO. DGMFN-3-Y-A2W-B2W-20



◆NOT AVAILABLE FOR SALE

⌘INCLUDED IN ADJ. SCREW KIT 942445
✤INCLUDED IN NEEDLE VALVE KIT 942447
*INCLUDED IN PILOT CHOKE F3 SEAL KIT 920084

MODEL CODE BREAKDOWN

(F3)-P DG5S-8-***-*-*-*-*-**-(*)-*-1*(LH)**

- (F3) MULTI-FLUID CAPABILITY (VITON SEALS)
- P** ELECTRICAL PLUG FEATURE (SEE NOTE BELOW)
- DG5S DIRECTIONAL CONTROL SUBPLATE MOUNTED SOLENOID CONTROLLED PILOT OPERATED RATED PRESSURE 210 bar (3000 PSI)
- 8 NFPA D06 (ISO-4401-8)
- SPOOL TYPE
- VALVE TYPE
  - A - SPRING OFFSET
  - B - SPRING CENTERED SINGLE SOLENOID
  - C - SPRING CENTERED
  - N - DETENT TYPE
- X - FAST RESPONSE (OMIT FOR STD. MODELS)
- SPOOL CONTROL MODIFICATION (OMIT WHEN NOT REQUIRED)
  - 1 - STROKE ADJUSTMENTS
  - 2 - PILOT CHOKE ADJUSTMENTS
  - 3 - PILOT CHOKE & STROKE ADJS.
  - 7 - STROKE ADJ. CYL. 'A' END ONLY
  - 8 - STROKE ADJ. CYL. 'B' END ONLY
- E - EXTERNAL PILOT PRESSURE (OMIT FOR INTERNAL PILOT PRESS)
- T - INTERNAL PILOT DRAIN (OMIT FOR EXTERNAL PILOT DRAIN)
- CHECK VALVE IN PRESSURE PORT (OMIT WHEN NOT REQUIRED)
  - K - 0.35 bar (5 PSI)
  - R - 3.45 bar (50 PSI)
  - S - 5.2 bar (75 PSI)
- ELECTRICAL CONNECTION OPTIONS
  - W - ½ NPT THREAD
  - WL - ½ NPT THREAD WITH SOLENOID INDICATOR LIGHTS.
- 5 - SERVICEABLE CORE TUBE S/A (OMIT FOR NON-SERVICEABLE)
- COIL IDENTIFICATION LETTER
- DESIGN -10/-11
- LEFT HAND ASSEMBLY
  - A - SPRING OFFSET
  - B - SPRING CENTERED SINGLE SOLENOID
  - (OMIT FOR R.H. ASS'Y)

ELECTRICAL PLUG FEATURE (OMIT IF NOT REQUIRED)

PA - INSTA-PLUG, MALE RECEPTACLE ONLY PB - INSTA-PLUG, MALE & FEMALE RECEPTACLE	PA3 (3 PIN) & PA5 (5 PIN) RECEPTACLES THAT MEET NFPA HYDRAULIC VALVE ELECTRICAL STANDARD T3. 5. 29M-1980.

To insure sustained efficiency and maximum trouble free life of this precision equipment, initial and continuous full flow filtration of the fluid medium is essential. Select and apply filters from the Vickers OFP, OFR, and OFRS series, which are available in 3, 10, and 25 micrometre filtration ratings.

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VICKERS

-LABORATORY WORK SHEET-

SOLENOID CONTROLLED, PILOT OPERATED DIRECTIONAL VALVES (DG5S-8-**-10)

Name:____________________

Reference: Service & Parts - I-3439-S
Industrial Products Catalog

I. PRELIMINARY OBSERVATION (SEE NAME PLATE ON SIDE OF 2-STAGE VALVE)

A. Model number of 2-stage valve:______________________________

B. Model number of pilot valve: ______________________________

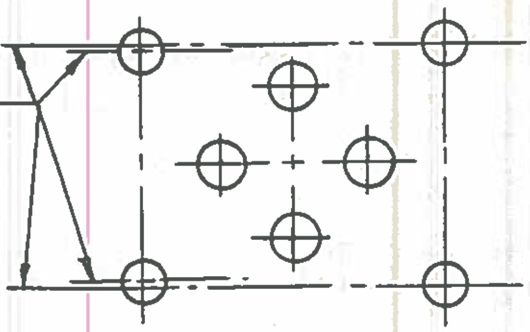
C. From the model code breakdown - I-3439-S, and visual observation, determine whether the following standard or optional items are included in subject valve:

1. Solenoid indicator lights Yes___ No___ Identified by: ______________
2. Insta-plug Yes___ No___ Identified by: ______________
3. Detented Yes___ No___ Identified by: ______________
4. Pilot choke adjustments Yes___ No___ Identified by: ______________
5. Check valve Yes___ No___ Identified by: ______________
6. Stroke adjustments Yes___ No___ Identified by: ______________
7. Fast response Yes___ No___ Identified by: ______________
8. Manual override Yes___ No___
9. Pilot pressure Internal___ External___ Identified by______________
10. Pilot drain Internal___ External___ Identified by______________
11. Main spool center condition (open, closed, etc..) ____________________ Identified by:______________________

II. PILOT VALVE DISASSEMBLY

A. Remove name plate. If indicator lights are provided, what purpose do they serve?____
____

B. Remove (4) pilot valve mounting screws with 4mm hex key and separate valve bodies. Do not lose or damage "O" ring seals.



Note difference in mounting hole centerlines.

C. Solenoid coil(s) may now be removed.
CAUTION: Static friction of "O" ring must be overcome by slightly rotating back and forth while pulling on solenoid.

Solenoid wires should be guided from terminal box during solenoid removal.

D. Remove 4 - 8mm hex head screws and retaining plate.

CAUTION: Internal parts are retained by core tube, so care must be taken to avoid dropping or damaging these parts during removal of core tube.

E. Core tube may now be removed

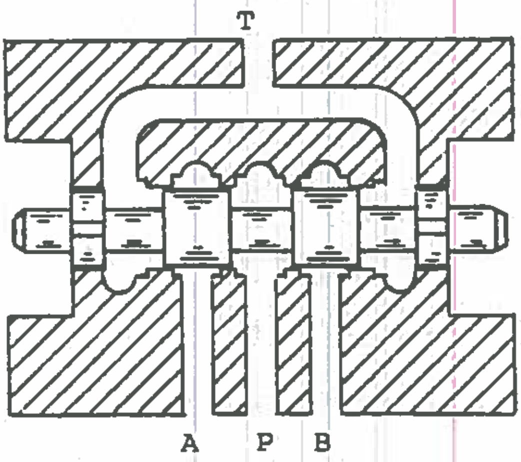
NOTE: Static friction of "O" ring and back-up ring must be overcome by rotating back and forth while pulling on core tube.

F. Remove spring and washer (if any).

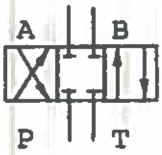
G. If double solenoid valve, remove other solenoid and associated parts by repeating steps C, D, E and F.

H. Spool may now be removed by gently pulling or pushing while rotating spool back and forth. DO NOT FORCE.

I. Determine spool type number by placing spool directly on sketch of body bore below. Type number:_______________ (See CTS-115)



Example shown is closed center type "2".



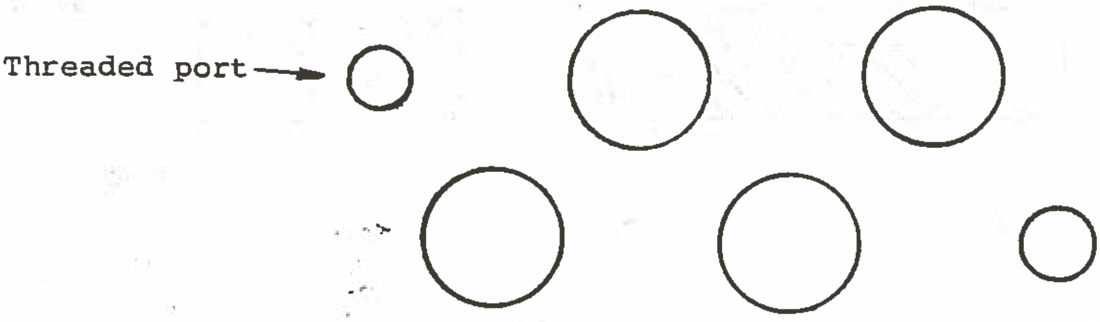
J. Valve is: (Check appropriate item)

1. Spring centered ______
2. Spring offset ______
3. Spring centered - single solenoid ______
4. Detented ______

K. How is above operator identified in model number? ______________________________

III. MAIN STAGE DISASSEMBLY

A. Turn main body (3/8-24 UNF Str. Thd. cylinder port gauge connections up) so mounting surface is visible. Identify and label ports.



B. Remove main valve body end covers and "O" rings (use 8mm hex key).

C. Remove springs, washers and main spool.

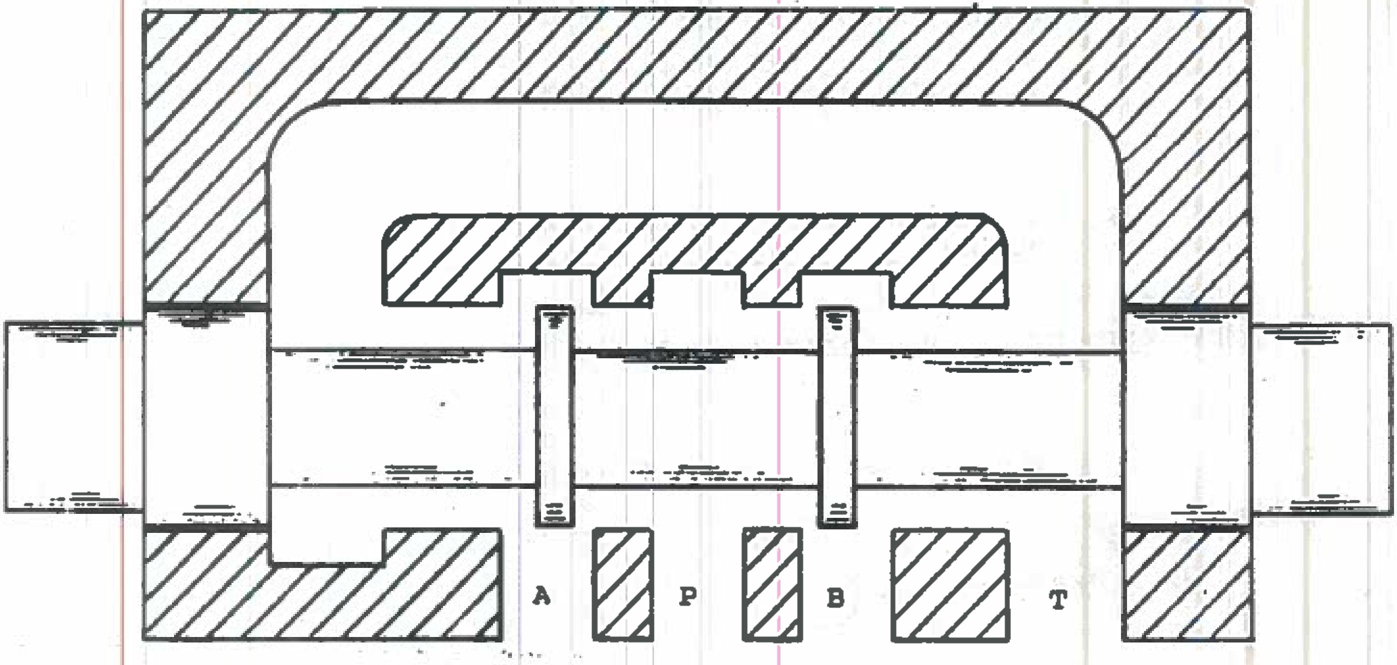
D. How is main valve centered? ______________________________

E. Is valve assembled for internal or external pilot pressure? ______________________________

F. Is valve assembled for internal or external drain?
____

G. What prevents the springs from direct contact with the spool? ______________________________

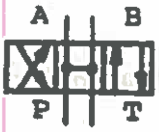
H. Determine spool type number by placing spool directly on sketch of body bore on next page. Type Number: ______________________________ (See CTS-115)



DG3S-8 BODY (MAIN STAGE)

Example:

P - Open
A - Open
B - Open
T - Open
Open Center
Type "O"



IV. CHECK DISASSEMBLY WITH INSTRUCTOR

V. REASSEMBLE

A. Reassemble valve as indicated by model number.

NOTE: Core tube retaining plate must be assembled with the chamfered edge toward body to avoid possible distortion of core tube when torquing 8mm hex head screws. Also note that the armature must be installed in the core tube with the recessed end turned toward the manual override button. If the armature is installed in the core tube backwards the spool will not shift when the solenoid is energized.

VI. SHOW THE GRAPHIC REPRESENTATION OF THE VALVE

Draw complete symbol showing both the solenoid controlled pilot stage and the pilot operated main stage.

PLUG INSTALLATION TABLE			
MODEL	"A" PLUG	"B" PLUG	"C" PLUG
DG5S-8-**-10	DOES NOT EXIST	*IN	OUT
DG5S-8-**-E-10		IN	*IN
DG5S-8-**X-10		OUT	OUT
DG5S-8-**X-E-10		IN	

* - Denotes Orifice Plug

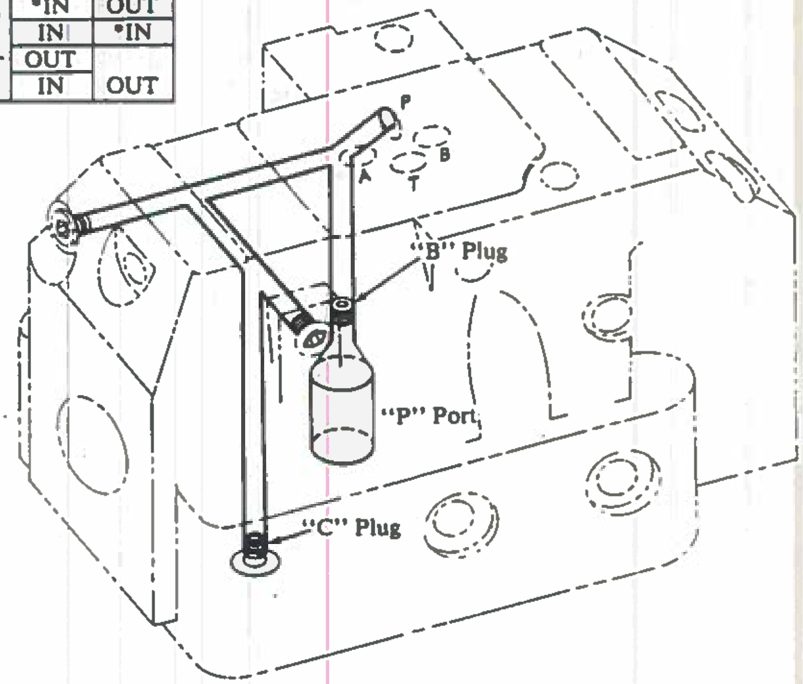


Figure 27a. DG5S-8-*-10 or DG5S-H8-*-20 Internal Pressure Passages & Plug Locations

---

DG5S8-10 (with or w/o check valve)
or DG5SH8-20 (with or w/o check valve)
Internal Drain Passages & Plug Locations

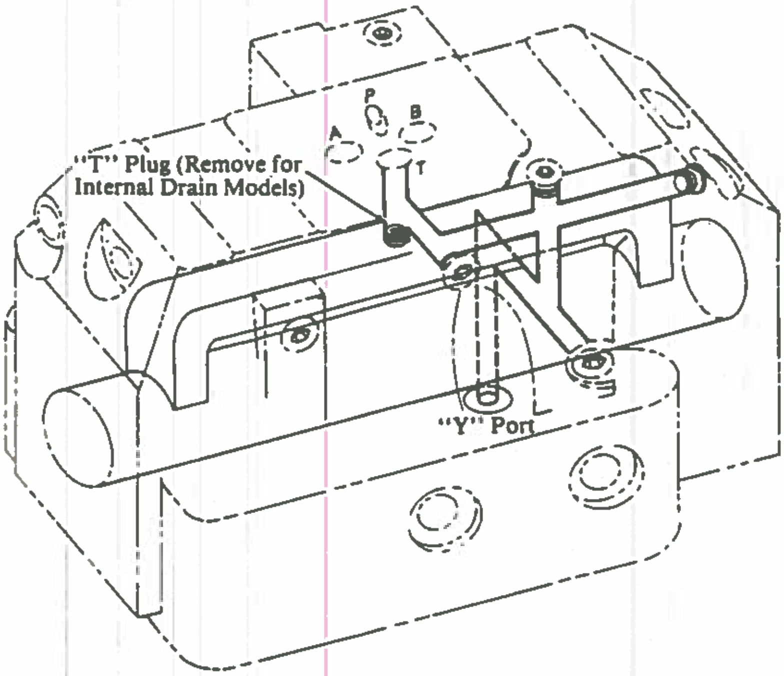
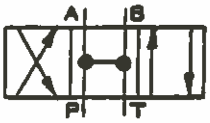


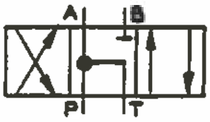
Figure 27c. DG5S8-**-10 or DG5S-H8-**-20
Internal Drain Passages & Plug Locations

DIRECTIONAL VALVE SPOOL TYPES

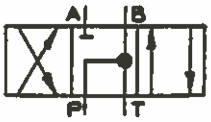
TYPE "0" ..... OPEN CENTER: All ports interconnected when spool is in neutral. Pump is unloaded. For use when there are no external forces acting on the cylinder when the spool is in neutral.



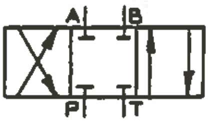
TYPE "1" ..... Allows flow from pressure "P" and cylinder "A" ports to tank. Can be used to support a load at cylinder "B" while pump is unloaded to tank with valve centered. Non symetrical spool.



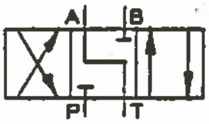
TYPE "11" .... Type 1 spool installed in reverse.



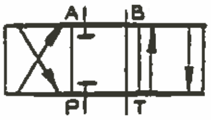
TYPE "2" ..... Closed Center: All ports are blocked when spool is in neutral. Prevents load from moving in either direction. Allows pump to be used for other operations.



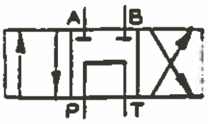
TYPE "3" ..... All ports blocked except cylinder "A" which connects to tank when spool is in neutral. Occasionally when both cylinder ports are blocked, there is a tendency for pressure to intensify at the rod end of a double acting single ended cylinder due to leakage. Type "3" spool prevents this by opening the rod end to tank. Non symetrical spool.



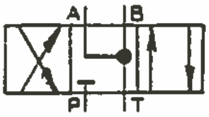
TYPE "31" .... Type 3 spool installed in reverse



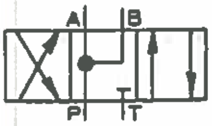
TYPE "4" ..... Tandem Center: Pressure to tank and both cylinders blocked in neutral with closed cross-over. Blocked cylinder ports prevent pressure build-up in the cylinder during series operation. Closed cross-over permits supporting a load while the spool is shifting.



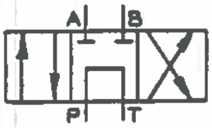
TYPE "6" .... Pressure blocked - both cylinders connected to tank in neutral, pemits free movement of the cylinder or motor when spool is in neutral. Allows the pump to be used for other operations. Pilot spools on spring centered pilot operated valves are Type 6 to permit main spool to center.



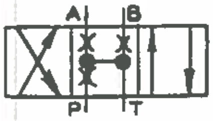
TYPE "7" .... Pressure open to both cylinders with tank blocked in neutral. Permits regenerative advance of cylinder with spool in neutral. Pressure is open to both ends of cylinder so greater area at cap end causes it to move forward.



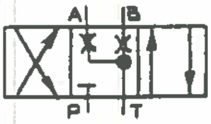
TYPE "8" .... Tandem Center: Pressure to tank and both cylinders blocked in neutral with open crossover. Same as Type 4, except it has "open crossover" and will not support a load while the spool is shifting.



TYPE "9" .... Restricted flow from all ports to tank in neutral. It is usually used as a two-position valve to provide smooth reversals of reciprocating cylinders.

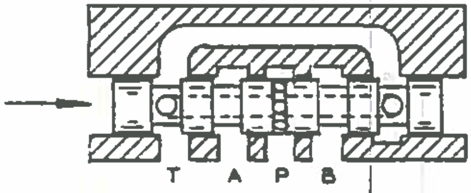


TYPE "33" .... Pressure blocked - restricted flow from both cylinder ports to tank in neutral. Used to prevent inadvertent drift of a horizontal cylinder by preventing pressure increase at the cylinder ports due to leakage.

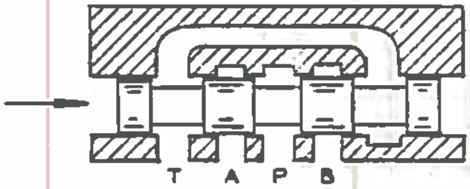


NOTE:

The graphic symbol for Types 4 and 8 spools depict parallel flow paths adjacent to pressure connection, while all other types show crossed flow paths adjacent to the pressure connection. This difference occurs due to the necessity of using a 3 land spool for a Type 4 and 8, basically, a 2 land spool for the other types.



3 LAND SPOOL



2 LAND SPOOL

Example: Shifting both spools to the right, we find flow out of "A" in a 3 land spool and flow out of the "B" using a 2 land spool.

VICKERS

ELECTRONIC CONTROL SYSTEMS

NEWSBYTES

March, 1985

④

This issue of Newsbytes provides a review of proportional and servo valves, to shed light on which is the proper product to use.

First, let's take a look at what a system designer must consider before taking a look at each type of valve.

Regardless of the system's complexity (i.e. the proposed application) the following points must be considered:

- Repeatable accuracy
- Dynamic response
- Stability
- Physical constraints, such as weight and size
- Fluid contamination
- Cost

Of these five categories, repeatable accuracy is the one most people will consider first. It is very important that the repeatable accuracy be well defined, as it will play an important role in selecting which product to use. Repeatable accuracy is especially important in positional control systems.

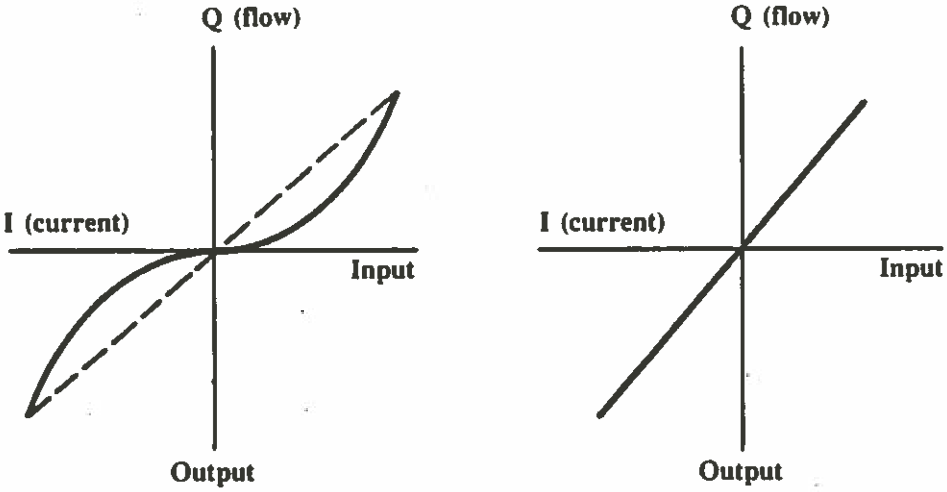
There are five characteristics of an electrohydraulic valve, such as a proportional or servo valve, which affect repeatable accuracy:

- Null flow gain
- Hysteresis
- Null bias
- Threshold (response sensitivity)
- Pressure gain

Both proportional and servo valves continuously convert an electrical input signal into a smooth and proportional hydraulic output signal. And, both units are controlled by electronic circuitry which is matched to the valve in question.

The term "proportional valve" is generously applied to pressure, flow and directional control valves which continuously convert a variable input signal into a smooth and proportional hydraulic output signal. In this respect, it can be regarded as an "umbrella" term which may also apply to servo valves. However, whenever proportional valves are mentioned in connection with hydraulic engineering, they only refer to those valves with a proportional characteristic which is quite distinct from servo valves.

Criterion	Proportional Valve	Servo Valve
Electrical Input	Proportional electro-magnetic with a relatively high power requirement, approximately 50 watts. Directly drives valve spool and compression springs.	Proportional electro-magnetic signal to the torque motor with relatively low power requirement, approximately .1 to .3 watts. Output power of the pilot stage is approximately 100 watts.
Static characteristic: Linearity between input and output	Valve operates at a low pressure drop, typically 120 psi. Resistances in the valve body have an effect on the linearity (saturation) when the through flow is large.	Valve operates with a high pressure drop, typically 1000 psi. Consequently, flow resistances within the valve body have little or no effect on linearity.



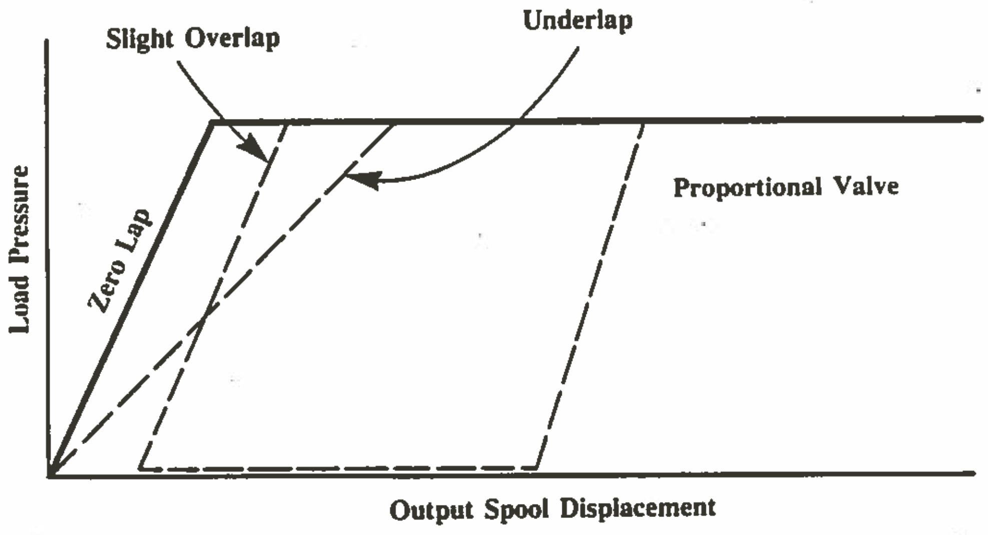
Hysteresis	Typically 3 to 8% Can be improved to approximately .5% with position control (electrical feedback) of the main stage.	Typically 1% Can be improved to approximately .1% with electrical position control of the main stage.

The figures for reversal error and response sensitivity are in the same order of magnitude.

Criterion	Proportional Valve	Servo Valve
Spool Lap	Typically ±20% overlap. This results in dead zones around the null position. However, it allows the use of conventional components with large production tolerances.	Typically 0, can be provided with over or under lap to satisfy application requirements. Manufacturing tolerances are exact, i.e., very close, and is normally a prerequisite for "closed-loop" control systems.

Spool lap has a direct effect on pressure gain and threshold. Pressure gain is higher for zero-lap and slightly overlapped spools, than the underlapped spools. Pressure lap is lower for underlap spools because the underlap produces an intermediate pressure to the cylinder ports at null. Full system pressure is not developed on a control port until the leakage path to return is blocked–requiring longer spool stroke than in an overlapped or zero-lap valve.

An overlapped spool exhibits a pressure gain as high as a zero-lap once the "dead zone" is traversed. Because fluid is somewhat compressible, a valve with high pressure gain can allow the system to respond rapidly to load disturbances. This allows for high accuracy under changing load conditions.



Criterion	Proportional Valve	Servo Valve
Dynamic Characteristics		
Response time for one complete stroke.	Approximately 40 and 60 ms	Approximately 5 to 20 ms
Frequency response, i.e., the frequency at which the output is 90° behind the input.	Typically 10 Hz	Typically 120 Hz. Because of their nature, servo valves can be "calibrated" for higher and lower frequency response to suit application requirements.

Criterion	Proportional Valve	Servo Valve
Filtration (recommended)	Typically 25 micron	Typically 25 micron
	The topic of required filtration has long been a controversial issue, especially regarding servo valves. the terminology of micron is really meaningless when considering hydraulic filtration. The proposed ISO standard more clearly defines the desired filtration level for long life expectancy of the product.	
	Today's state-of-the-art servo valves have been engineered to operate well on contamination levels found acceptable for conventional hydraulic components.	
Electronics	Both products can be applied using "off-the-shelf" amplifiers and power supplies, or with electronic "modules" specifically designed to meet the application requirements.	
	Normally open-loop, requiring no feedback transducer.	Normally closed-loop, requiring a feedback transducer.
Areas of Application	Normally in an open-loop control circuit where load regulation is not critical. Example: Position control where load must be positioned to an accuracy of ±.125 inch or greater.	Normally in a closed-loop control circuit where load regulation is critical. Example: Position control where load must maintain a regulated position of ±.002 inch or better.
	Can be applied in closed-loop applications where load regulation is not critical. Example: Position control where load position must be ±.050 inch or greater.	Can be applied in an open-loop application with overlapped spool and load regulation is not critical. Example: Position control where load position must be ±.010 inch or greater.
	Accuracy cannot be taken as an absolute number as there are many factors outside the performance of the valve which will determine the overall system performance. Each application must be considered on its own merits.	
Cost	The problems encountered in electrohydraulic control systems are due to the complexity involved. Their design places high requirements on the conception, commissioning and troubleshooting of such systems. the use of a proportional valve in a closed-loop circuit is often considered due to its reasonable price. Such a valve can only be used in such applications when the requirements placed on the control system are not exacting, and on a closer observation, it often turns out to be rather problematical (dead zones, dynamics, etc.).	
	The proportional valve can be used to advantage in the following areas: • Electrical remote control for continuous and smooth variation of pressure and volumetric flow. • Large number of fixed-program pressure and flow-control valves can be replaced by a single proportional valve. • Acceleration and deceleration of cylinders and hydraulic motors.	
Physical Constraints	Typically:	Typically:
Weight:	5 gpm, 6 pounds 10 gpm, 19 pounds 20 gpm, 19 pounds	5 gpm, 2 pounds 10 gpm, 2 pounds 20 gpm, 3 pounds
Size:	5 gpm: 11" long × 2" wide × 4" high 10 gpm: 14" long × 3" wide × 5" high 20 gpm: 14" long × 3" wide × 5" high	5 gpm: 4" long × 2" wide × 3" high 10 gpm: 4" long × 2" wide × 3" high 20 gpm: 4" long × 4" wide × 3" high

Printed in U.S.A.

6

PRESSURE CONTROLS

MODEL SERIES CGE - 06/10 20 DESIGN

REMOTE ELECTRICALLY MODULATED RELIEF VALVES

GENERAL DATA

These Vickers valves provide the capability to modulate system pressure using a remote electrical controller. The pressure setting of the valve is approximately proportional to input current; increasing current provides increasing pressure.

APPLICATION GUIDANCE

Response Time

For response time, see chart. Note that response time will decrease with increasing flows. Also response time is dependent on the amount of all volume in the pressure side of the circuit; less oil volume under pressure will result in faster response times.

Step input currents of less magnitude will decrease response time proportionally. Decreasing pressure response time will be approximately 1/4 of the above values.

RATINGS

Pressure and Flow .... See specifications charts.

Pressure Change Due to Temperature:

The maximum pressure deviation between 26.7°C (80°F) and 65.6°C (150°F) is 4.8 bar (70 PSI).

Hysteresis:

5 percent of maximum current (without dither - see curves).

3 percent of maximum current (with dither).

Recommended Dither ....10 mA (RMS) at 60 Hz

Power Required (Maximum) .... 5.75 Watts

Nominal Coil Resistance ....18 Ohms

Current Required ....0-500 mA dc (Not polarity sensitive)

PRESSURE AND FLOW RATINGS

Model Number	Maximum Pressure		Maximum Flow	
	bar	PSI	Liters/Min.	GPM
CGE-06-1-2•	70	1000	170	45
CGE-06-3•-2•	210	3000	170	45
CGE-10-1-2•	70	1000	380	100
CGE-10-3•-2•	210	3000	380	100

For minimum pressure see chart below and curves on pages 4 and 5.

MINIMUM CONTROLLABLE FLOWS

Model Number	Maximum Controllable Flow	
	Liters/Min.	GPM
CGE-06-•-2•	7.6	2
CGE-10-•-2•	11.4	3

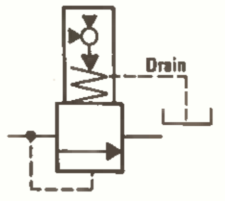
MINIMUM PRESSURE SPECIFICATIONS (ZERO CURRENT)

Model	Flow		Minimum Pressure	
	Liters/Min.	GPM	bar	PSI
CGE-06-1-2•	76	20	5.2	75
CGE-06-3-2•			10.4	150
CGE-06-3V-2•			20.7	300
CGE-10-1-2•	189	50	5.2	75
CGE-10-3-2•			10.4	150
CGE-10-3V-2•			20.7	300

See curves for other typical values.

RESPONSE TIME, TYPICAL

Model Number	Step Input mA	Flow		Pressure Increase		Response Time-ms
		Liters/Min.	GPM	bar	PSI	
CGE-06-1-2•	0-500	76	20	5- 70	75-1000	250
CGE-06-3•-2•				10-210	150-3000	290
CGE-10-1-2•		189	50	5- 70	75-1000	400
CGE-10-3•-2•				10-210	150-3000	240



STANDARD GRAPHICAL SYMBOL FOR FLUID POWER DIAGRAMS

MODEL CODE

C G E - 06 - 3 (V) - 2*

1 2 3 4 5 6 7

1 Relief Valve

2 Mounting Type

G - Manifold or subplate.

3 Remote Electrically Modulated

4 Valve Size

06 - 60 series (3/4" nominal size).

10 - 100 series, 1-1/4" nominal size.

5 Pressure Rating (Maximum)

1 - 70 bar (1000 PSI)

3 - 210 bar (3000 PSI)

6 High Vent Spring

(For 3000 PSI pressure models only). Omit if not required.

7 Design Number

Subject to change. Installation dimensions remain as shown for design numbers 20 through 29.

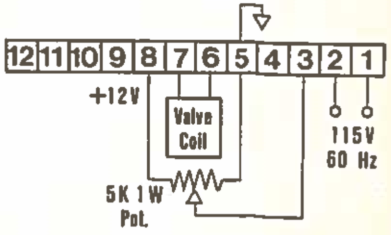
Manual Operation

The manual override adjustment operates over the full pressure range and should be set just above the maximum pressure to be controlled electrically. For example, set the override at 2200 PSI when the maximum electrically controlled pressure is 2000 PSI.

The override adjustment can also be used for complete manual operation of the valve during set up and troubleshooting by isolating the pilot head section from the manual override section. This is done by removing an access plug and reversing a button in the pilot section. (See diagram on page 511040-2.)

Electrical Power Supply

Vickers power supply EMCS-•-30 is recommended for controlling this valve. The basic wiring diagram is show. (Amplifier switch should be in position 1.) See drawing 521557 for complete description.



EMCS-*-30 POWER SUPPLY

VICKERS®

Vickers, Incorporated
A Libbey-Owens-Ford Company
Troy, MI 48084

INSTALLATION DATA | PRESSURE CONTROLS | MODEL SERIES CGE - 06/10 | CAPACITIES TO 100 GPM | PRESSURES TO 3000 PSI | FOR 3/4" & 1-1/4" PIPING | MANIFOLD OR SUBPLATE MOUNTING | DWG. NO. 511040

PRESSURE CONTROLS

Revised 10-85

INSTALLATION DATA

Filter Screen

This valve contains two filter screens which may require periodic cleaning depending on system cleanliness. One filter is located in the pressure port of the valve and is held in place with a snap ring.

The other filter screen is shown in Section Z-Z. To remove this filter, first shut down machine and remove the access plug, button and spool. Clean screen and reassemble as shown.

Tank Connection

The tank connection should be piped directly to tank through a surge free line to minimize back pressure. If tank line back pressure exceeds system pressure by 100 PSI a malfunction may occur. Any pressure in the tank line is additive when the valve is controlled by the manually adjusted (non-electrical) head.

Drain Connections

Drain line must be full size unrestricted and connected directly to reservoir so that it terminates below reservoir fluid level. No other lines are to be connected to this drain line. Pressure in this line (taken at the valve) not to exceed 15 PSI.

SUBPLATES AND BOLT KITS

Valves, subplates and bolt kits must be ordered separately.

Example:
One (1) CGE-06-3-2* Valve
One (1) CGM-06-20 Subplate
One (1) BKCG06-604 Bolt Kit †
or
One (1) CGE-10-3-2* Valve
One (1) CGM-10-20 Subplate
One (1) BKCG10-616 Bolt Kit †

Note

To replace a conventional CG or CG5 valve, install the CGE valve on the existing mounting, and connect the pilot drain (in the control head) to tank.

Mounting Bolt Torque (Maximum Recommended)

CGE - 60 Series	47 Nm (35 lb ft)
CGE - 100 Series	95 Nm (70 lb ft)

When subplate is not used, a machined pad (as indicated by subplate shaded area) must be provided for mounting; pad must be flat within 0.01270 millimeters (.0005 in.) and smooth within 1.6 micrometer (63 microinch). Mounting bolts when provided by customer, should be SAE grade 7, or better.

FILTRATION

For satisfactory service life of these components in industrial applications, use full flow filtration to provide fluid which meets ISO cleanliness code 16/13 or cleaner. Selections from Vickers OFP, OFR and OFRS series are recommended. Refer to Section "P"-"Filters" and Section "S"-"Technical Information" of Vickers Catalog 400, or contact your Vickers representative for further filtration advice.

FLUIDS AND SEALS

Petroleum base fluid with viscosity range from 32 cSt to 50 cSt at 38°C (150 SSU to 225 SSU at 100°F). Refer to Vickers data sheet I-286-S, "Hydraulic Fluid and Temperature Recommendations for Industrial Machinery" (included in the Technical Information section of Vickers Catalog 400) for hydraulic fluid and temperature recommendations.

Temperature Range of Oil

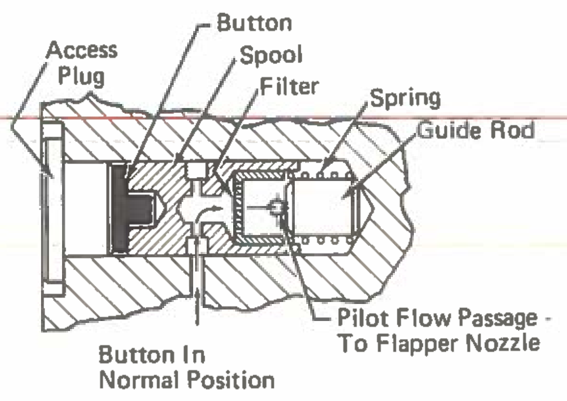
The recommended oil temperature is from 38°C to 54°C (100°F to 130°F). The valve will function acceptable from 21°C to 65°C (70°F to 150°F).

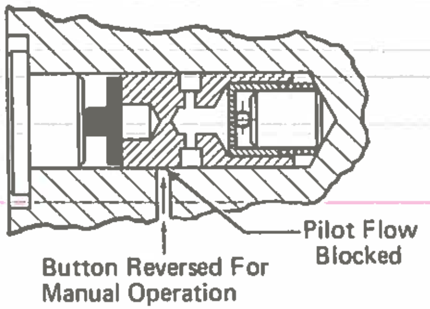
WEIGHT

CGE-06	22 lbs.
CGE-10	29 lbs.

SERVICE INFORMATION

Refer to specific Vickers part drawing or overhaul manual for service information. A complete "Index to Industrial Products Service Information" is included in the Technical Information section of Vickers Catalog 400.


Access Plug
Button
Spool
Filter
Spring
Guide Rod
Pilot Flow Passage - To Flapper Nozzle
Button In Normal Position



SECTION Z-Z

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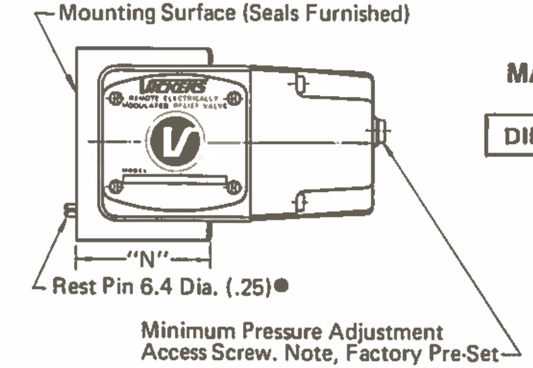
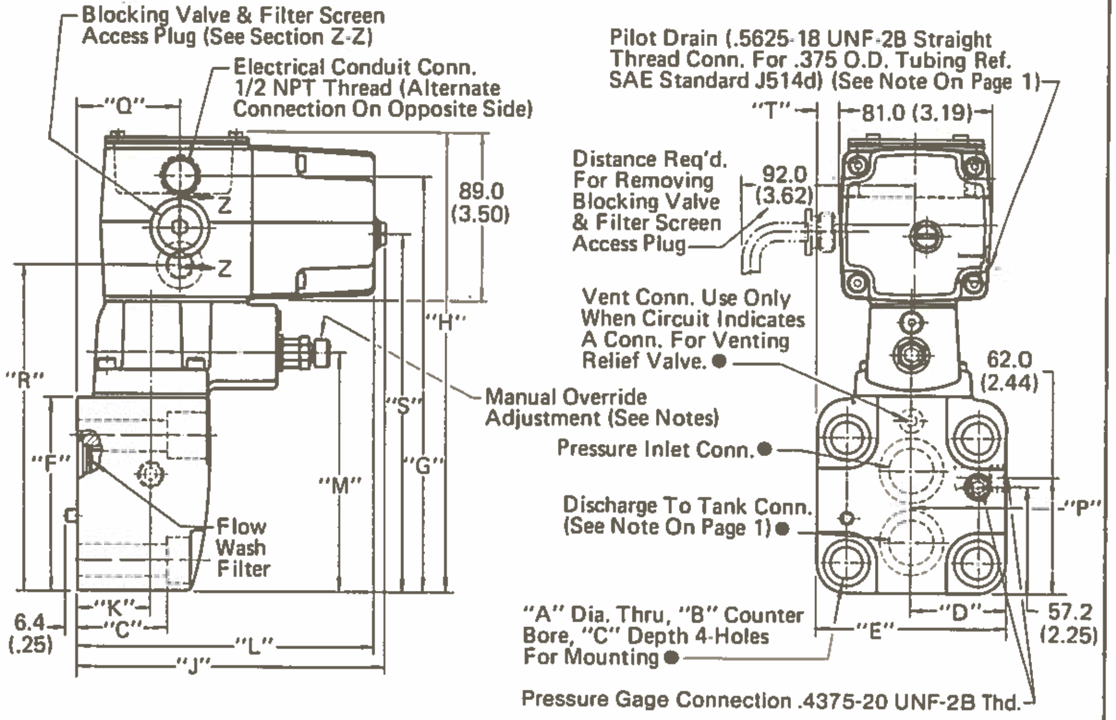
† See "Bolt Kit" information in Section "Q" of Vickers Catalog 400, for bolt thread size and length.

MODEL SERIES CGE-06/10-**-2* MANIFOLD OR SUBPLATE MOUNTING

DIMENSIONS SHOWN MILLIMETERS (INCHES)

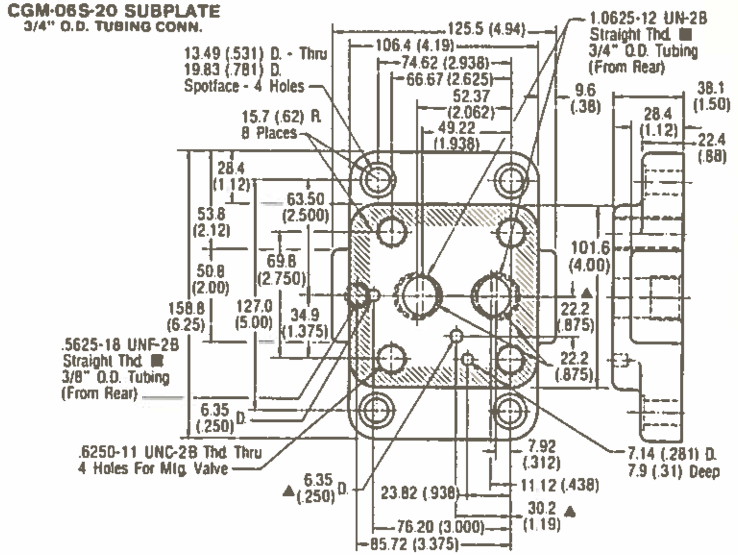

Mounting Surface (Seals Furnished)
VICKERS
REMOTE ELECTRICALLY MODULATED RELIEF VALVE
"N"
Rest Pin 6.4 Dia. (.25)●
Minimum Pressure Adjustment Access Screw. Note, Factory Pre-Set


Blocking Valve & Filter Screen Access Plug (See Section Z-Z)
Electrical Conduit Conn. 1/2 NPT Thread (Alternate Connection On Opposite Side)
"Q"
Z
Z
89.0 (3.50)
"H"
"R"
"S"
"F"
"M"
"G"
Flow Wash Filter
6.4 (.25)
"K"
"C"
"L"
"J"
Manual Override Adjustment (See Notes)
Pilot Drain (.5625-18 UNF-2B Straight Thread Conn. For .375 O.D. Tubing Ref. SAE Standard J514d) (See Note On Page 1)
"T"
81.0 (3.19)
Distance Req'd. For Removing Blocking Valve & Filter Screen Access Plug
92.0 (3.62)
Vent Conn. Use Only When Circuit Indicates A Conn. For Venting Relief Valve.●
62.0 (2.44)
Pressure Inlet Conn.●
Discharge To Tank Conn. (See Note On Page 1)●
"P"
"D"
"E"
57.2 (2.25)
"A" Dia. Thru, "B" Counter Bore, "C" Depth 4-Holes For Mounting●
Pressure Gage Connection .4375-20 UNF-2B Thd.

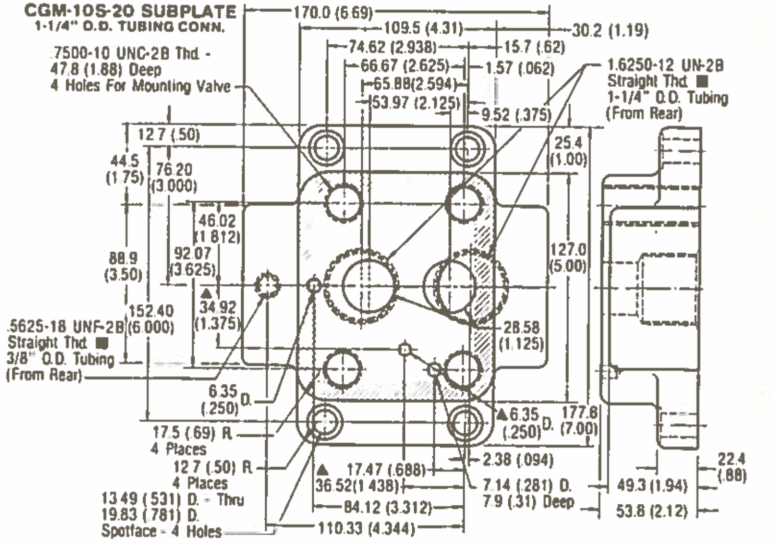
● See Mounting Subplates For Locating Dimensions.

MOUNTING SUBPLATES

CGM-06S-20 SUBPLATE
3/4" O.D. TUBING CONN.



CGM-10S-20 SUBPLATE
1-1/4" O.D. TUBING CONN.



▲ Drain Connection - Required For Sequence ("Y") Models Only - Must Be Machined By User.
■ Ref. SAE Standard J514

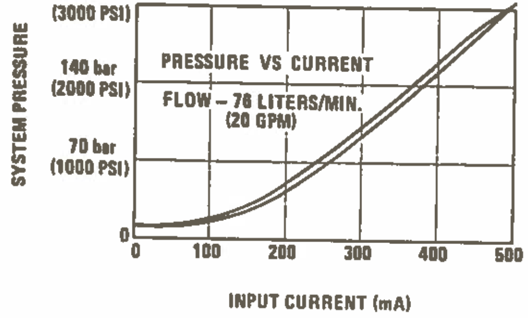
DIMENSIONS

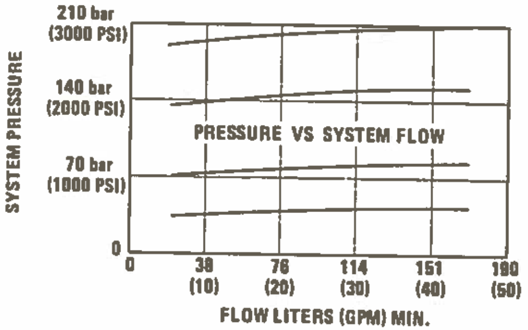
Model Series Number	A	B	C	D	E	F	G	H	J	K	L	M	N	P	Q	R	S	T	Weight (Approx.)	
																			kg	Lbs.
CGE-06-***-2*	16.65 (.656)	24.60 (.968)	47.8 (1.88)	53.4 (2.10)	106.4 (4.19)	103.1 (4.06)	220.5 (8.68)	244.4 (9.62)	165.0 (6.50)	38.9 (1.53)	158.6 (6.25)	127.0 (5.00)	71.4 (2.81)	35.1 (1.38)	54.1 (2.13)	173.0 (6.81)	192.8 (7.59)	13.3 (.53)	10	22
CGE-10-***-2*	19.85 (.781)	29.35 (1.156)	63.5 (2.50)	66.8 (2.63)	133.4 (5.25)	109.5 (4.31)	231.7 (9.12)	255.5 (10.06)	170.5 (6.71)	42.9 (1.69)	164.1 (6.46)	138.2 (5.44)	85.9 (3.38)	46.0 (1.81)	59.6 (2.35)	184.2 (7.25)	204.2 (8.04)	26.9 (1.06)	13	29

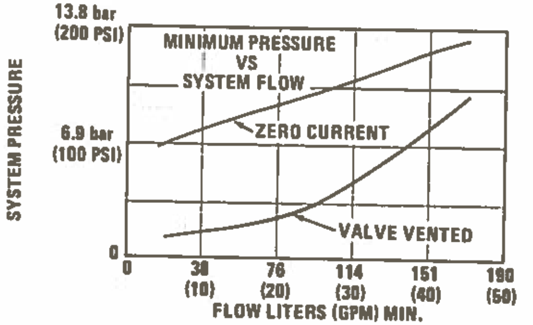
TYPICAL PERFORMANCE CURVES

The performance curves shown are for typical valves. Minor differences in these characteristics will be evident from valve to valve.

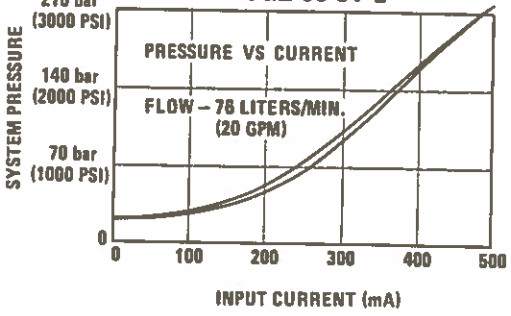
CGE-06-3-2*

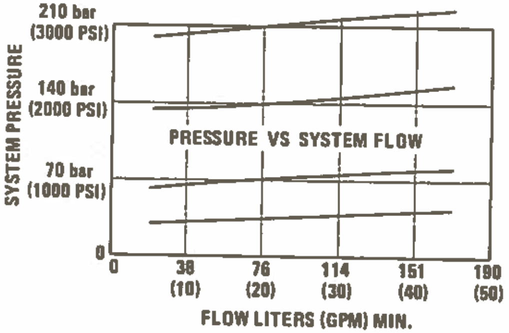


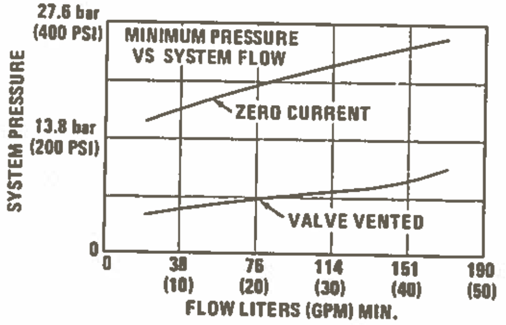




CGE-06-3V-2*







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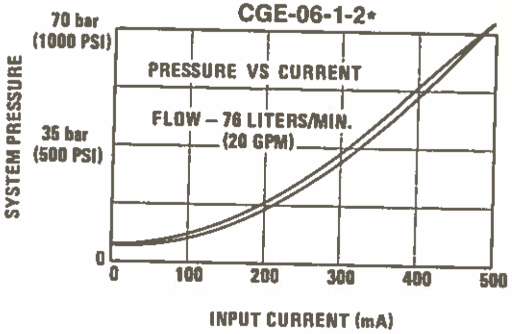
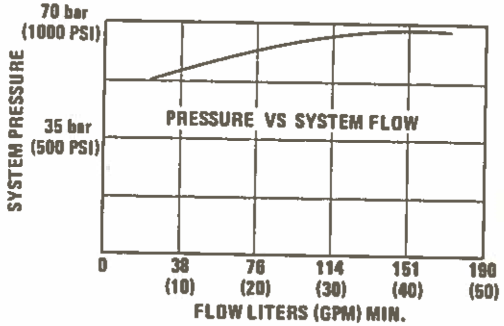
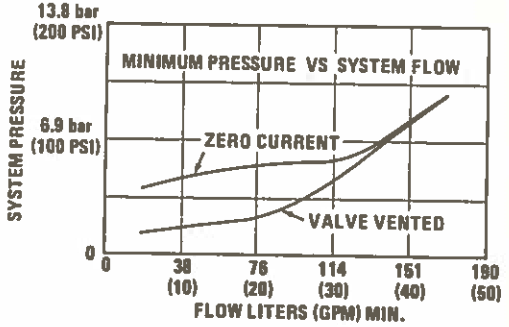
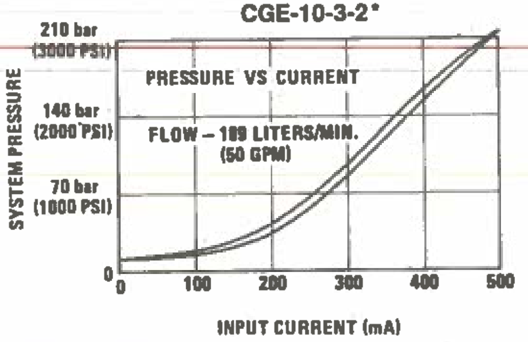


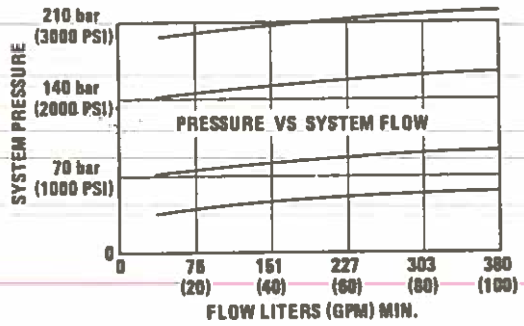


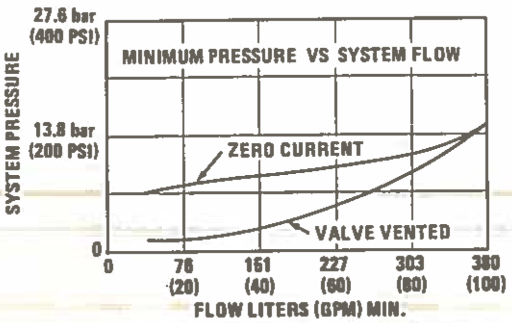


TYPICAL PERFORMANCE CURVES

CGE-10-3-2*







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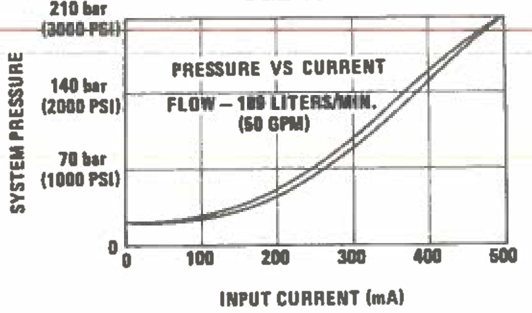

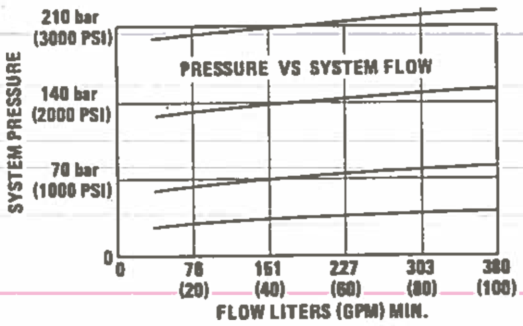

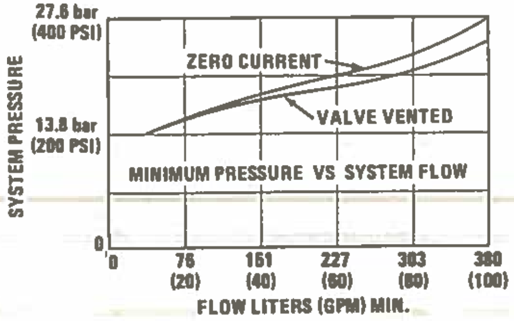

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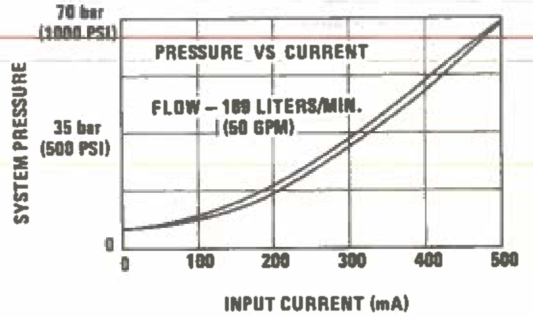

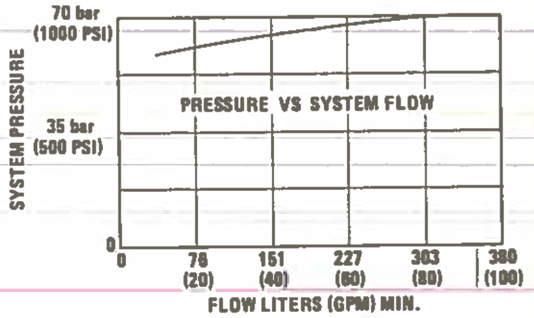

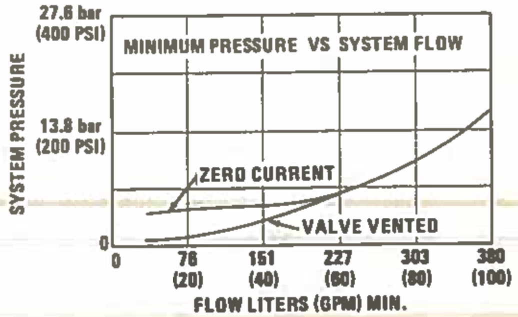

(6)

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FLOW CONTROLS

MODEL SERIES FGE - 06 1* DESIGN

REMOTE ELECTRICALLY MODULATED

GENERAL DATA

This valve provides output flow relative to an electrical input current. Precise output flow rates can be obtained by adjusting the magnitude of the input current.

Valve output flow rates can be manually controlled from an operators panel by setting a position on a potentiometer dial. Automatic load speed control can be accomplished by programming the magnitude of the electrical input signal to the valve.

This valve is used for speed control of hydraulic motors and cylinders. Typical applications are speed control of hydraulic spindles and axis carriages for machine tools; and speed control of injection cylinders or screws for plastic injection molding machinery.

The valve can be used in either an open loop or closed loop mode. The valve's electrical-hydraulic interface makes the unit ideally suited for applications requiring an electrical feedback loop of load speed to command signal.

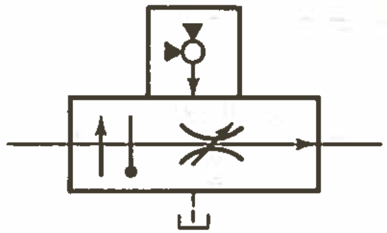
The valve is temperature and pressure compensated to reduce flow variation with changes in oil temperature and inlet or outlet pressure.

RATINGS

Maximum Flow Capacity:	
FGE-06-30-002-*-12	30 GPM
FGE-06-60-002-*-12	60 GPM
Minimum Flow Capacity	See chart.
Maximum Inlet Pressure	3000 PSI
Maximum Outlet ("C" Port) Pressure	2000 PSI
Minimum Pressure Differential Between Inlet and Outlet:	
FGE-06-30-002-*-12	150 PSI
FGE-06-60-002-*-12	200 PSI
Response Time (From Change in Current to Start of Flow Change)	50 ms
Shift Time (90 Percent of Change):	
Minimum to Full	.38 Sec.
Full to Minimum	.39 Sec.
Minimum Cycle Time (Without Amplitude Decline)	1.0 Sec.
Repeatability of Set Flow	±1 percent
Pressure Compensation	±5 percent
Resolution Percent Flow Setting	≤1 percent
Hysteresis:	
FGE-06-30-002-*-12	1 GPM
FGE-06-60-002-*-12	2 GPM
Temperature Drift:	
FGE-06-30-002-*-12	0.02 GPM/°F
FGE-06-60-002-*-12	0.05 GPM/°F
Temperature Range:	
Nominal	120°F
Maximum	150°F
Power Required for Rated Flow	3.2 watts (20Ω nom. coil resist.)
Current Required	0 to 400 mA

MINIMUM METERED FLOW CAPACITY

Valve Differential Pressure (PSI)	Minimum Outlet Flow (Cu. In./Min.)	Minimum Inlet Flow (Cu. In./Min.)	
		Internal Pilot	External Pilot
1000	35	85	35
1500	50	100	50
2000	75	125	75
2500	125	175	125
3000	200	250	200



STANDARD GRAPHICAL SYMBOL FOR FLUID POWER DIAGRAMS

MODEL CODE

F G E - 06 - 30 - 002 - (E) - 12

1 2 3 4 5 6 7 8

1 Flow Control

2 Mounting

Manifold or subplate.

3 Remote Electrically Modulated

4 Nominal Valve Size

06 - 3/4"

5 Maximum Flow Capacity

30 - 30 GPM
60 - 60 GPM

6 Coil Designation

Number times 10 equals coil resistance in ohms.

7 Pilot Supply

E - External
Blank - Omit for internal.

8 Design Number

Subject to change. Installation dimensions remain as shown for design numbers 10 through 19.

APPLICATION GUIDANCE

Flow Adjustment

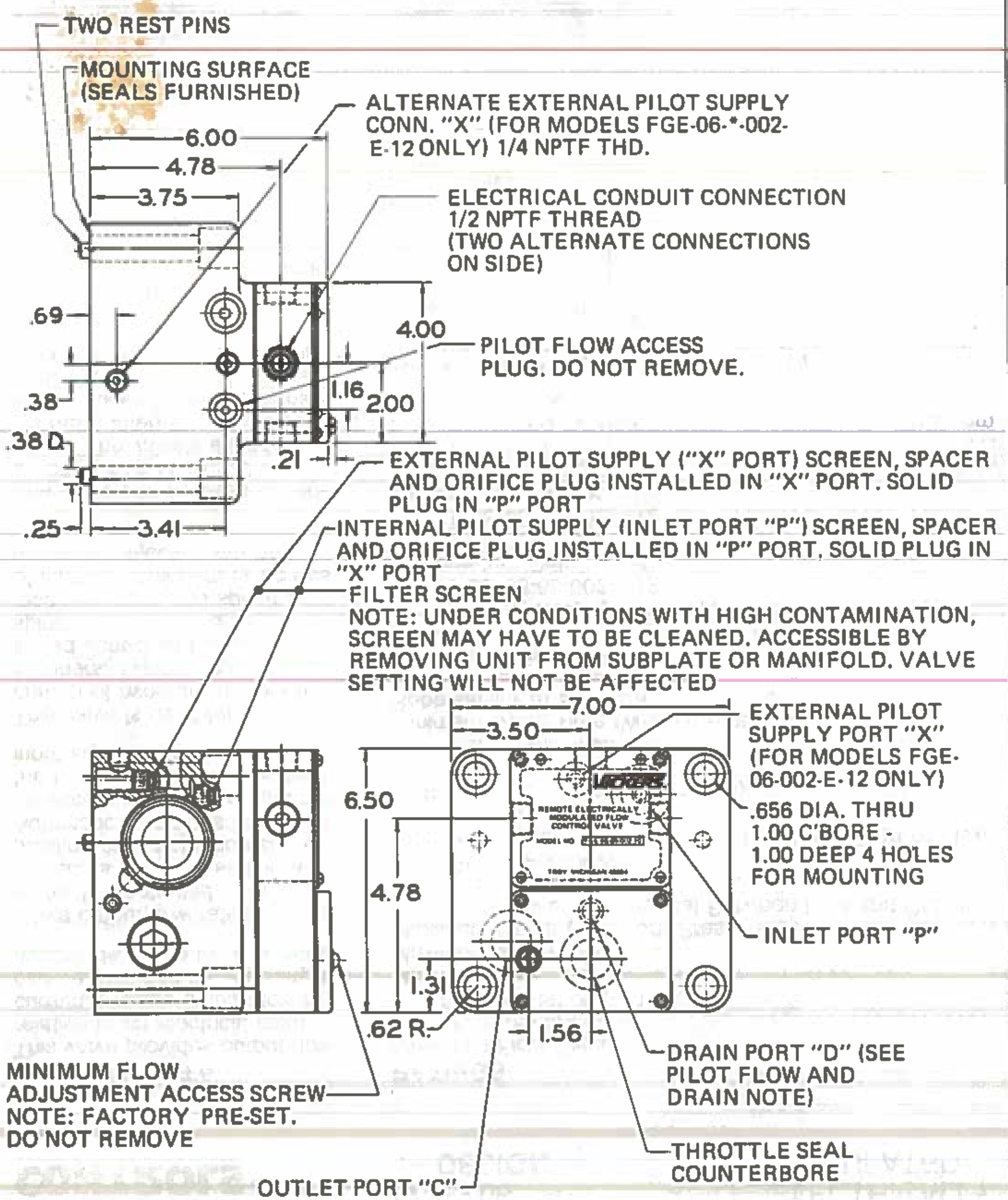
Adjust flow by applying regulated current to the coil. The Vickers power supply model EMCS-*-30 is recommended for use with this valve, however, customers may provide their own power.

In the event customers provide their own electrical power, the supply used should conform to the following specifications (deviations will result in reduced valve performance):

Output Current	0-400 mA D.C.
Current Regulation	±2 mA
Repeatability	±2 mA
Resolution	≤1 mA
Temperature Drift	≤0.10 mA/°F.
Response (90 Percent of Change)	≤6 ms
Dither (Optional)	0-25 mA RMS @ 49-61 Hz

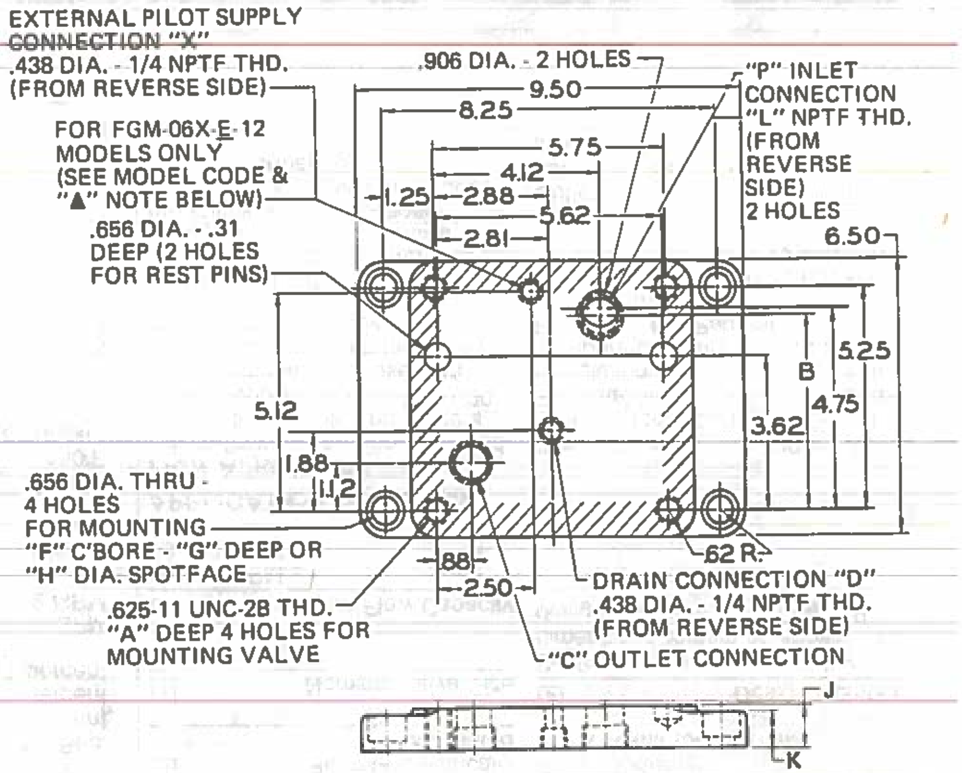
Note:
Outlet flow returns to minimum flow upon loss of electrical power.

FGE-06-*0-002-(E)-12 VALVE


TWO REST PINS
MOUNTING SURFACE (SEALS FURNISHED)
ALTERNATE EXTERNAL PILOT SUPPLY CONN. "X" (FOR MODELS FGE-06-*-002-E-12 ONLY) 1/4 NPTF THD.
6.00
4.78
3.75
ELECTRICAL CONDUIT CONNECTION 1/2 NPTF THREAD (TWO ALTERNATE CONNECTIONS ON SIDE)
.69
4.00
PILOT FLOW ACCESS PLUG. DO NOT REMOVE.
.38
1.16
2.00
.38 D
.21
EXTERNAL PILOT SUPPLY ("X" PORT) SCREEN, SPACER AND ORIFICE PLUG INSTALLED IN "X" PORT. SOLID PLUG IN "P" PORT
.25
3.41
INTERNAL PILOT SUPPLY (INLET PORT "P") SCREEN, SPACER AND ORIFICE PLUG INSTALLED IN "P" PORT. SOLID PLUG IN "X" PORT
FILTER SCREEN
NOTE: UNDER CONDITIONS WITH HIGH CONTAMINATION, SCREEN MAY HAVE TO BE CLEANED. ACCESSIBLE BY REMOVING UNIT FROM SUBPLATE OR MANIFOLD. VALVE SETTING WILL NOT BE AFFECTED
7.00
3.50
EXTERNAL PILOT SUPPLY PORT "X" (FOR MODELS FGE-06-002-E-12 ONLY)
6.50
.656 DIA. THRU 1.00 C'BORE - 1.00 DEEP 4 HOLES FOR MOUNTING
4.78
INLET PORT "P"
1.31
.62 R.
1.56
DRAIN PORT "D" (SEE PILOT FLOW AND DRAIN NOTE)
MINIMUM FLOW ADJUSTMENT ACCESS SCREW NOTE: FACTORY PRE-SET. DO NOT REMOVE
THROTTLE SEAL COUNTERBORE
OUTLET PORT "C"

For port, rest pin, and mounting locating dimensions refer to subplate.

MOUNTING SUBPLATES


EXTERNAL PILOT SUPPLY CONNECTION "X" .438 DIA. - 1/4 NPTF THD. (FROM REVERSE SIDE)
.906 DIA. - 2 HOLES
9.50
8.25
"P" INLET CONNECTION "L" NPTF THD. (FROM REVERSE SIDE) 2 HOLES
FOR FGM-06X-E-12 MODELS ONLY (SEE MODEL CODE & "▲" NOTE BELOW)
5.75
4.12
1.25
2.88
5.62
.656 DIA. - .31 DEEP (2 HOLES FOR REST PINS)
2.81
6.50
5.25
B
4.75
3.62
5.12
1.88
1.12
.656 DIA. THRU - 4 HOLES FOR MOUNTING "F" C'BORE - "G" DEEP OR "H" DIA. SPOTFACE
.62 R.
.88
2.50
DRAIN CONNECTION "D" .438 DIA. - 1/4 NPTF THD. (FROM REVERSE SIDE)
.625-11 UNC-2B THD. "A" DEEP 4 HOLES FOR MOUNTING VALVE
"C" OUTLET CONNECTION
J
K

Model Number	Dimensions							
	A	B	F	G	H	J	K	L
FGM-06-12	1.00	4.75	1.00	.62		1.00	.88	3/4
FGM-06X-12	1.06	4.62			1.00	1.69	.81	1

Note

"▲" for subplates with external pilot supply connection add "E" to model number. Example FGM-06X-E-12. Straight thread subplate available on request.

When subplate is not used, a machined pad (as indicated by subplate shaded area) must be provided for mounting; pad must be flat within .0005 inch and smooth within 63 microinch. Mounting bolts, when provided by customer, should be SAE grade 7, or better.

Pilot Flow
Pilot flow is 50 cu. in./min. Internal pilot supply (inlet port "P") screen, spacer and orifice plug is installed in "P" port. Solid plug in "X" port.

External pilot supply ("X" port) screen, spacer and orifice plug is installed in "X" port. Solid plug in "P" port. External pilot supply pressure must be 200 to 3000 PSIG and need not be constant.

Drain
Drain line must be full size unrestricted and connected directly to reservoir so that it terminates below the reservoir fluid level. No other lines are to be connected to this drain line. Pressure in this line (taken at the valve) not to exceed 15 PSI.

Dither
Dither is not required. A 60 Hz adjustable amplitude dither signal is available on the Vickers power unit model EMCS-•-30. The dither signal amplitude can be adjusted to optimize load performance but the maximum value should not exceed 25 mA.

Interchangeability
Model FGE-06-•0-002-•-12 can be mounted in place of model FG-06-••-1•. If model FGE-06-•0-002-E-12 is required the alternate external pilot supply connection "X" provided on the side of the valve body (1/4 NPTF Thd.) must be used.

SUBPLATES AND BOLT KITS

Valves, subplates and mounting bolts must be ordered separately.
Example:
One (1) FGE-06-60-002-12 Valve
One (1) FGM-06-10 Subplate or
One (1) BKFGE06-649 Bolt Kit †

Maximum Recommended Mounting Bolt Torque . . . . .125 lb. ft.

† See "Bolt Kit" information in section "Q" of Vickers Catalog 400 for bolt thread size and length.

FLUIDS AND SEALS

Petroleum base fluid with viscosity ranging between 150 SSU and 225 SSU at 100°F. Refer to Vickers data sheet I-286-S, "Hydraulic Fluid and Temperature Recommendations for Industrial Machinery" (included in the Technical Information section of Vickers Catalog 400) for hydraulic fluid and temperature recommendations.

FILTRATION

For satisfactory service life of these components in industrial applications, use full flow filtration to provide fluid which meets ISO cleanliness code 16/13 or cleaner. Selections from Vickers OFP, OFR and OFRS series are recommended. Refer to Section "P"-"Filters" and Section "S"-"Technical Information" of Vickers Catalog 400, or contact your Vickers representative for further filtration advice.

WEIGHT

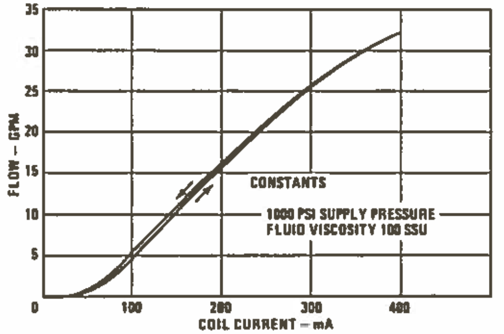
Valve . . . . 44 lbs.
Subplates:
FGM-06-•-12 . . . . . . 14 lbs.
FGM-06X-•-12 . . . . 21 lbs.

SERVICE INFORMATION

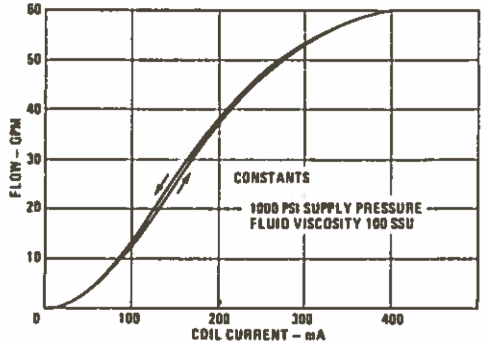
Refer to specific Vickers part drawing or overhaul manual for service information. A complete "Index to Industrial Products Service Information" is included in the Technical Information section of Vickers Catalog 400.

TYPICAL PERFORMANCE CURVES

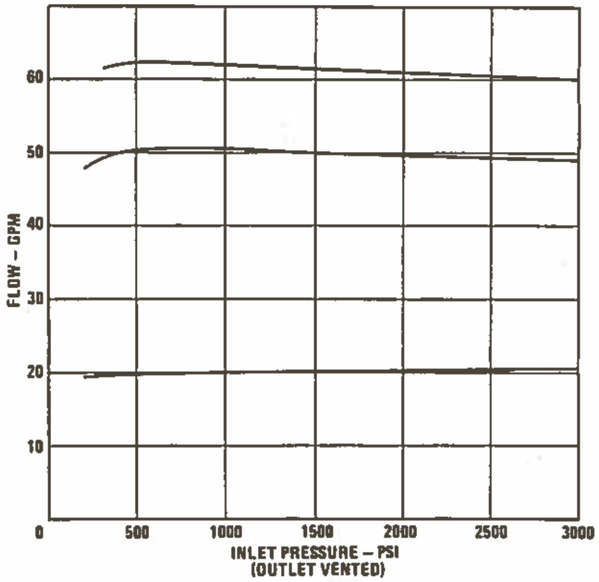
TYPICAL PERFORMANCE
FGE-06-30-002-•-12
OPEN LOOP

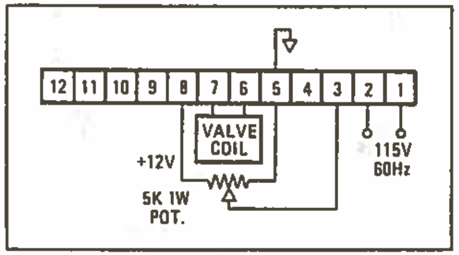


TYPICAL PERFORMANCE
FGE-06-60-002-•-12
OPEN LOOP



TYPICAL PRESSURE COMPENSATION
FGE-06-60-002-•-12





ELECTRICAL CONNECTIONS

Used with Vickers power supply EMCS-•-30.

Note
Valve coil is not polarity sensitive. Command potentiometer connection as shown CW rotation increases output. Command potentiometer is supplied with EMCS-P-30. See drawing 521557.

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ELECTROHYDRAULIC PROPORTIONAL SOLENOID VALVES

7a

MODEL SERIES K*G4V-3-*-*-*-*-40 NFPA-D01 (ISO-4401-03) INTERFACE

2-WAY THROTTLE TYPE (T) & 4-WAY DIRECTIONAL TYPE (D)

GENERAL DATA

The primary function of these proportional directional valves in a hydraulic circuit is to direct and meter fluid flow in proportion to the command signal to the valve. This, in turn, would determine the direction, velocity and acceleration/deceleration of a fluid cylinder, or the rotation of a fluid motor. These valves have wet armature type solenoids.

Port connections are made by mounting the valve on a subplate or manifold.

The electrical connections to the valve are made in the electrical wiring housing. A ground terminal is provided. Alternate connections complying with the German DIN standard 43650 are also available.

ELECTRONICS

Electronic controllers matched to the K*G4V-3 valve are available to provide optimum performance from the valves. For full details on these controllers, see catalog drawing 521575.

RATINGS

Performance:

Maximum Flow (Each Flow Path) ..... See chart on page 519806-4
Rated Flow @ 10 bar (145) PSI Δ P ... See chart on page 519806-4
Maximum Operating Pressure .... 350 bar (5000 PSI)
Maximum Tank Line Operating Pressure (See $\boxed{11}$) ..... 155 bar (2250 PSI)
Maximum Recommended Pressure Drop .... 105 bar (1500 PSI)

When used as other than a normal 4-way valve or other than is shown on page 519806-2, consult your local Vickers representative.

Frequency Response (−3 db) .... 8 Hz @ 45° Phase Lag
Hysteresis .... ±4 percent
Repeatability .... ±1 percent
Dead Band (Overlap) .... 15-35 percent of Full Solenoid Input
Vickers electronic controller has a dead band eliminator to reduce this value to zero.
Production Tolerance (Valve To Valve) .... ±10 percent

Note:
All performance values were obtained using Vickers standard electronic controller.

Typical Step Response Time For Spring Centered Models (@ Rated Flow and Pressure):

0-100 percent .... 100 ms (Center spool position to full spool travel)
100-0 percent .... 40 ms (Full spool travel to center spool position - fast drop out)
10-90 percent ...70ms (10 percent of full flow to 90 percent of full flow)
90-10 percent ..30 ms (90 percent of full flow to 10 percent of full flow)
100-100 percent .... 110 ms (100 percent of full spool travel in one direction to 100 percent of full spool travel in the reverse direction)

Controller Model	Solenoid Voltage (DC) Rating	Connections
EM-VT-12-10	12	Terminal Strip
EM-VT-24-10	24	
EM-VP-12-10	12	DIN Standard - 41612 Type "F32"
EM-VP-24-10	24	

MODEL CODE

K * G 4 V - 3 - ** (*) (L) ** S - M (P**) - * - * - 40 - (S***)
$\boxed{1}$ $\boxed{2}$ $\boxed{3}$ $\boxed{4}$ $\boxed{5}$ $\boxed{6}$ $\boxed{7}$ $\boxed{8}$ $\boxed{9}$ $\boxed{10}$ $\boxed{11}$ $\boxed{12}$ $\boxed{13}$ $\boxed{14}$ $\boxed{15}$ $\boxed{16}$ $\boxed{17}$

$\boxed{1}$ **Proportional**

$\boxed{2}$ **Function**
D - Directional control valve (double solenoid models).
T - Throttle valve (single solenoid models).

$\boxed{3}$ **Mounting Type**
Manifold or Subplate

$\boxed{4}$ **Solenoid Operated**

$\boxed{5}$ **Rated Pressure**
350 bar (5000 PSI)

$\boxed{6}$ **Interface**
NFPA D01 (ISO-4401-03)

$\boxed{7}$ **Spool Type (Center Condition)**
0 - Open Center, All Ports
2 - Closed Center, All Ports

$\boxed{8}$ **Spring Arrangement**
B - Spring centered; single solenoid.
BP - Spring centered; single solenoid with manual operator in end cap.
C - Spring centered; double solenoid.
F - Spring offset; shift to center; single solenoid.
FP - Spring offset; shift to center; single solenoid with manual operator in end cap.

$\boxed{9}$ **Left Hand Assembly**
For single solenoid models only. Omit for standard right hand assembly.

$\boxed{10}$ **Flow Rating**
@ 10 bar (145 PSI) Pressure Drop
Example - 08 - 8 L/Min (2 GPM)
15 - 15 L/Min (4 GPM)
19 - 19 L/Min (5 GPM)

$\boxed{11}$ **Standard Metering**

$\boxed{12}$ **Flag Symbol**
(Separating electrical from mechanical features)
M - Electrical features & options

$\boxed{13}$ **Electrical Plug Features**
"PB" - Insta-plug male & female receptacle.
"PA3" - NFPA 3-pin conduit connector (single solenoid models only).
"PA5" - NFPA 5-pin conduit connector.

$\boxed{14}$ **Electrical Options**
"U" - German DIN standard 43650 (plug on each solenoid).
"W" - 1/2 NPT thread wiring port in wiring box.

$\boxed{15}$ **Coil Identification Letter**
"DC" voltage only.
Indicate other voltages and frequencies by suitable letter (see chart on page 519806-2).

$\boxed{16}$ **Design Number**
Subject to change. Installation dimensions remain as shown for design numbers 40 through 49.

$\boxed{17}$ **Special Features**
"S440" - Special spool type. (See graphic symbol on page 519806-2).

VICKERS®
Vickers, Incorporated
A Libbey-Owens-Ford Company
Troy, MI 48084

INSTALLATION DATA

PROPORATIONAL DIRECTIONAL & THROTTLE CONTROLS

MODEL SERIES KDG4V-3-**-*-40 NFPA-D01 (ISO-4401-03) INTERFACE

TO 38 L/MIN. (10 GPM)

TO 350 BAR (5000 PSI)

MANIFOLD OR SUBPLATE MOUNTING

DWG. NO. 519806

PROPORTIONAL VALVES

K

Released 8 - 85

DESIGN AND CONSTRUCTION

Proportional solenoid valves function much like conventional on-off solenoid valves, except that when a pulse width modulated input signal is applied, the spool can be infinitely positioned to achieve throttling (restriction) of the fluid flow as well as directional control. This ability to control flow and direction in a single control valve can also accomplish acceleration and deceleration control, when using the controller ramp function or ramped input command.

The pulse width modulated control regulates the valve spool position either side of center. Primary usage is in open loop velocity control systems. However, closed loop position control is possible, although the accuracy achievable is less than provided by servo valves.

More accurate pressure compensated flow control in open loop systems is possible by applying a hydrostat in conjunction with the proportional valve.

The KDG4V-3 valve is adaptable to mobile vehicle and industrial machinery applications.

APPLICATION GUIDANCE

Metering on double solenoid spring centered models is either from the "A" port to the tank port or from "B" to "T". The "P" to "A" and "P" to "B" flow paths function like a conventional on-off solenoid valve. That is, they do not meter.

The two-way, single solenoid, spring centered version is a throttle valve that can be single ported for 7.5 GPM ("B" to "T" or "A" to "P") flow or double ported (dual flow path) for 15 GPM. An -S440 option is available allowing meter-in and meter-out control for a single acting cylinder.

Typical circuits are shown that portray meter-out control using either the single solenoid or the double solenoid configuration.

A method of using that single solenoid, spring centered version as a meter-in control is also shown. Metering is from "B" to "P". Metering-in from "A" to "T" is not recommended due to the tank port pressure limitation.

OPERATING DATA

Electrical Information

Solenoids are identified by a letter in the model number.

Example:
KDG4V-3-2C15S-M-W-H-40

Solenoid Voltage (DC) Rating	Coil Identification Letter	amp	watt
12	G▲	2.6	31
24	H▲	1.3	31

Note:
Solenoids are designed to function continuously at ±10 percent of the rated voltage.

▲ See DIN standard solenoids information on page 3.

Note:
On all models when solenoid "A" is energized, flow is always P → A. When solenoid "B" is energized, flow is always P → B. Solenoid "A" & "B" are identified on the diagram plate on the side of the valve.

③ Electrical Connection Note:

1/2 NPT thread for KDG4V-3 models. The wiring box is a NEMA type 4 enclosure. German DIN standard 43650 connections are also available. The conduit connection readily adapts to connector-receptacle assemblies on the market.

Lead Wire Insulation

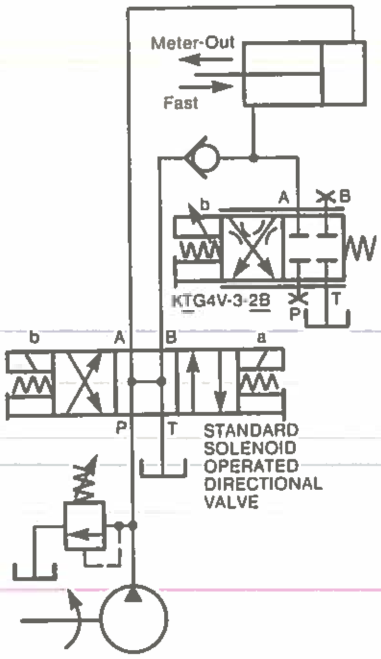
Class "H" on both ac and dc solenoid lead wires.

Solenoid Indicator Lights:

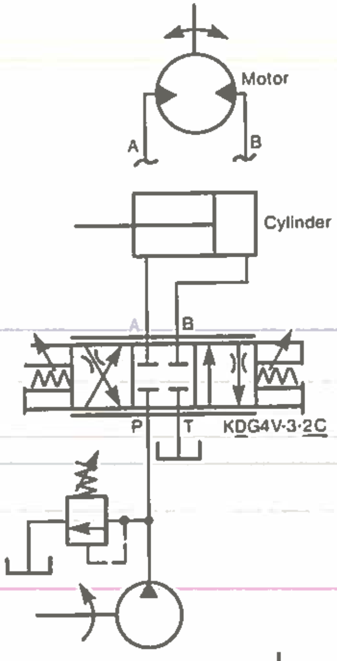
Not available.

STANDARD GRAPHICAL SYMBOLS FOR FLUID POWER DIAGRAMS

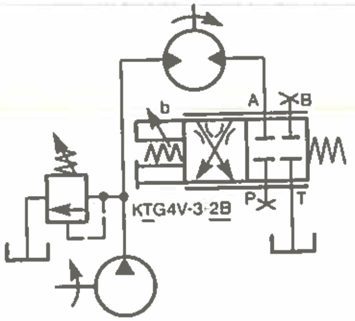
CYLINDER METER-OUT (ROD EXTEND) FAST RESPONSE (ROD RETRACT)



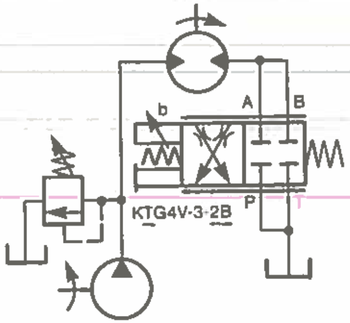
MOTOR OR CYLINDER METER-OUT EITHER DIRECTION



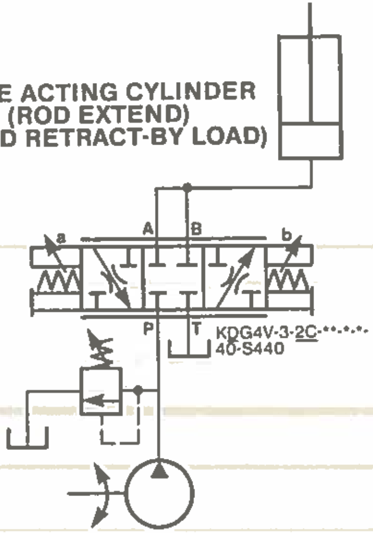
MOTOR DRIVE SINGLE FLOW PATH METER-OUT



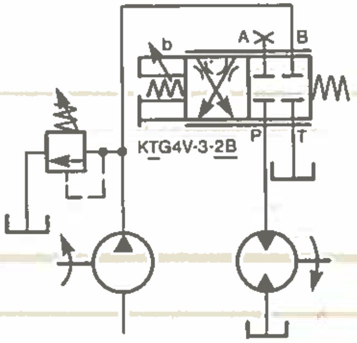
MOTOR DRIVE DUAL FLOW PATH METER-OUT



RAM OR SINGLE ACTING CYLINDER METER-IN (ROD EXTEND) METER-OUT (ROD RETRACT-BY LOAD)

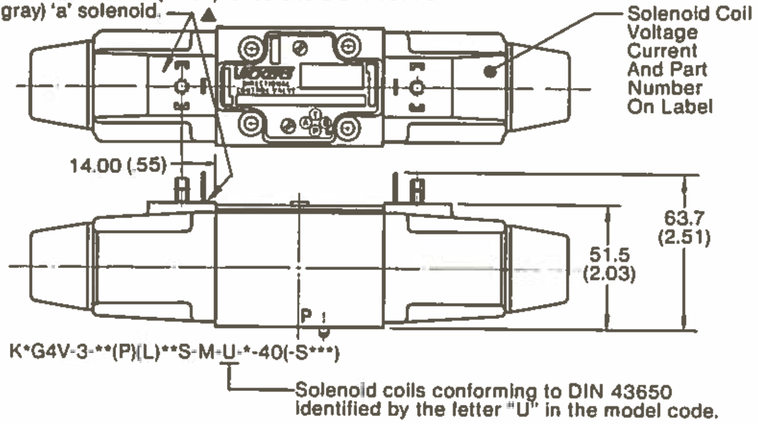


MOTOR DRIVE SINGLE FLOW PATH METER-IN

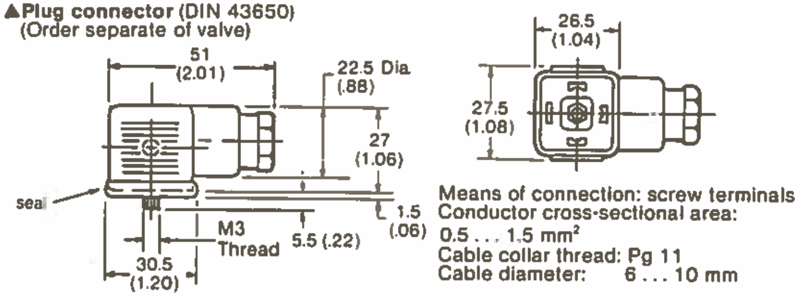


GERMAN DIN STANDARD 43650 PLUG

This interface is used for connecting electrical receptacles conforming to Germin DIN Standard 43650. Mating receptacles are available. Order Part No. 710775 (black) 'b' solenoid or 710776 (gray) 'a' solenoid. ▲


Solenoid Coil Voltage Current And Part Number On Label
14.00 (.55)
63.7 (2.51)
51.5 (2.03)
P
K*G4V-3-**(P)(L)**S-M-U-*-40(-S***)
Solenoid coils conforming to DIN 43650 identified by the letter "U" in the model code.

▲Plug connector (DIN 43650)
(Order separate of valve)



Means of connection: screw terminals
Conductor cross-sectional area: 0.5 . . . 1.5 mm²
Cable collar thread: Pg 11
Cable diameter: 6 . . . 10 mm

INSTA-PLUG FEATURE

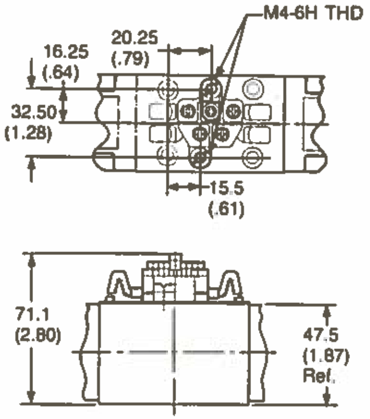
INSTA-PLUG FEATURE

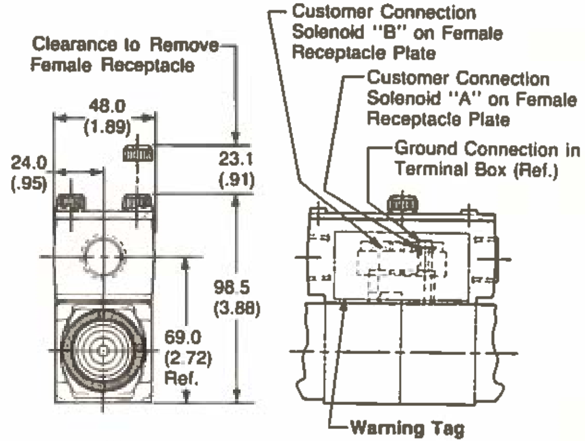
Vickers "Insta-Plug" provides a means of disconnecting electrical power to the valve without actually breaking individual wire connections. The male half of the plug is attached to the top of the valve body. Solenoid leads are connected to this half of the plug by Vickers. The mating plug is attached inside the wire housing. Terminals are provided on top of it for the convenience of connecting machine wires.

Two thumb screws are used to hold the wiring housing with the female plug half to the valve.

These screws are captive to avoid loss on servicing. To disconnect the valve they are loosened until clear, allowing the wiring housing to be pulled away from the valve body; thereby disconnecting electrical power to the valve. Note that the ground or fifth post is longer than the other four, providing a first-make/last-break ground feature.

KDG4V-3-2C15S-M**PB**-W-G-40: The **PB** configuration includes both the male and female (retained in the wiring housing) halves for a complete plug-in unit.




Clearance to Remove Female Receptacle
48.0 (1.89)
24.0 (.95)
23.1 (.91)
98.5 (3.88)
69.0 (2.72) Ref.
Customer Connection Solenoid "B" on Female Receptacle Plate
Customer Connection Solenoid "A" on Female Receptacle Plate
Ground Connection in Terminal Box (Ref.)
Warning Tag

"Electrical Power Must Be Disconnected Before Removing or Replacing This Receptacle".

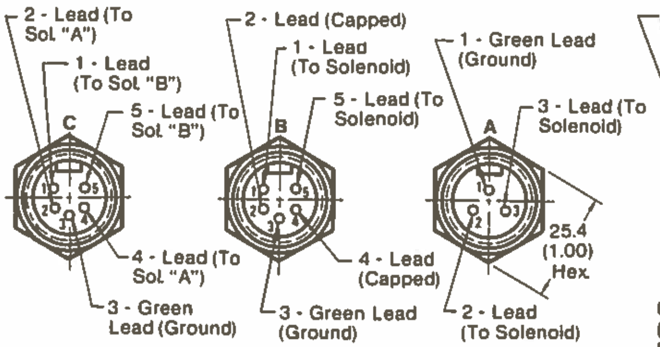
NFPA HYDRAULIC VALVE ELECTRICAL CONNECTOR - STANDARD T3.5.29-1980

The receptacle is a standard three or five pole "mini-change" electrical connector with shortened leads and terminals added. The five pole has four leads 101.6mm (4.0") long and one 177.8mm (7.0") long. The No. 4 and No. 2 leads are attached to the "A" solenoid and the No. 5 and No. 1 leads are attached to the "B" solenoid. The three pole plug has two leads 101.6mm (4.0") long and one 177.8mm (7.0") long. The No. 2 and No. 3 leads are attached to the solenoid. All of the wires have underwriters recognized non-solder insulated eyelet terminals. The green wire is used for the ground connection (No. 8 screw furnished).

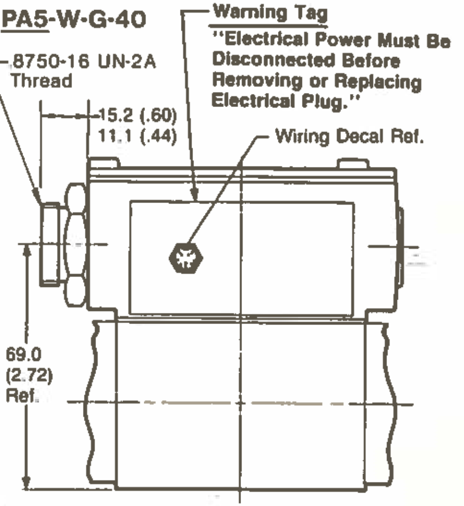
Receptacle will be prewired to the solenoid eyelets. The connection will be made via No. 6 screws and nuts insulated with black electrical tape.

Electrical Rating 600 volts 3 pole, 10 amps and 5 pole, 8 amps.

TYPICAL MODEL: KDG4V-3-2C15S-M**PA5**-W-G-40

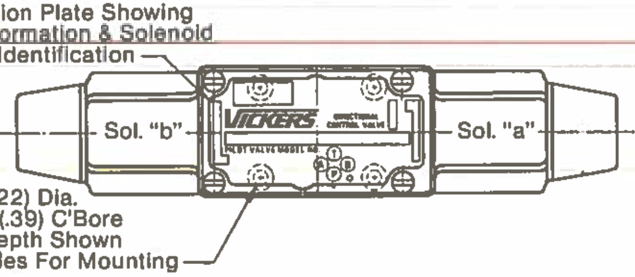


A -3 Pole Single Solenoid Models
B -5 Pole Single Solenoid Models
C -5 Pole Double Solenoid Models


Warning Tag
"Electrical Power Must Be Disconnected Before Removing or Replacing Electrical Plug."
Wiring Decal Ref.
.8750-16 UN-2A Thread
15.2 (.60)
11.1 (.44)
69.0 (2.72) Ref.

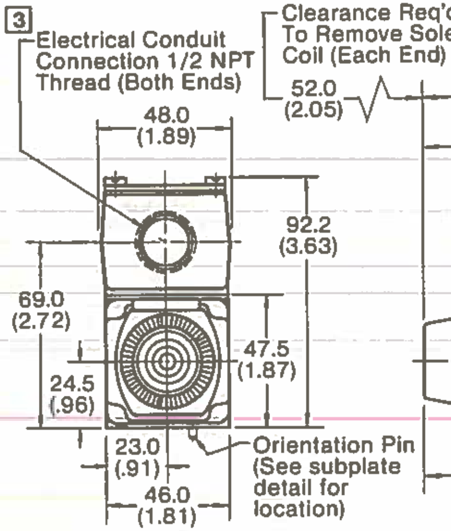
DOUBLE SOLENOID MODELS, SPRING CENTERED
KDG4V-3-*C**S-M-W-*-40

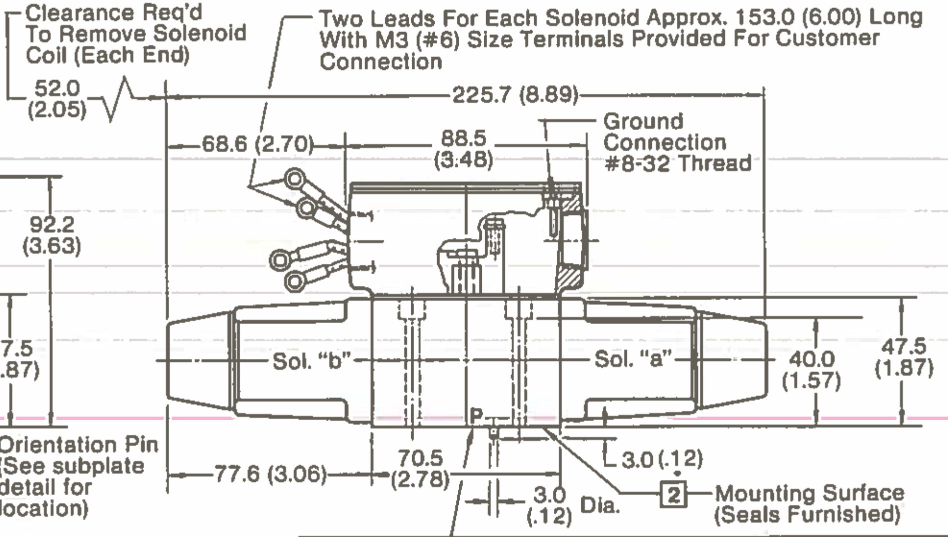
Spring Centered Models - When solenoid is de-energized the spool is returned to the spring centered position. (Right hand assemblies shown.)



☐ REFER TO NOTES ON THIS OR OTHER PAGES.

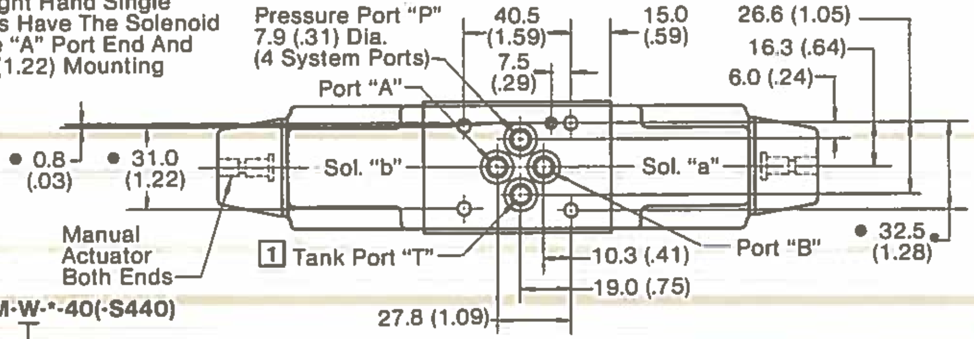
DIMENSIONS SHOWN MILLIMETERS (INCHES)


3 Electrical Conduit Connection 1/2 NPT Thread (Both Ends)
48.0
(1.89)
92.2
(3.63)
69.0
(2.72)
47.5
(1.87)
24.5
(.96)
23.0
(.91)
46.0
(1.81)
Orientation Pin (See subplate detail for location)


Clearance Req'd To Remove Solenoid Coil (Each End)
52.0
(2.05)
Two Leads For Each Solenoid Approx. 153.0 (6.00) Long With M3 (#6) Size Terminals Provided For Customer Connection
225.7 (8.89)
68.6 (2.70)
88.5
(3.48)
Ground Connection #8-32 Thread
Sol. "b"
Sol. "a"
P
40.0
(1.57)
47.5
(1.87)
77.6 (3.06)
70.5
(2.78)
3.0 (.12)
3.0
(.12) Dia.
2 Mounting Surface (Seals Furnished)

BODY STAMPED TO IDENTIFY PRESSURE PORT SIDE.

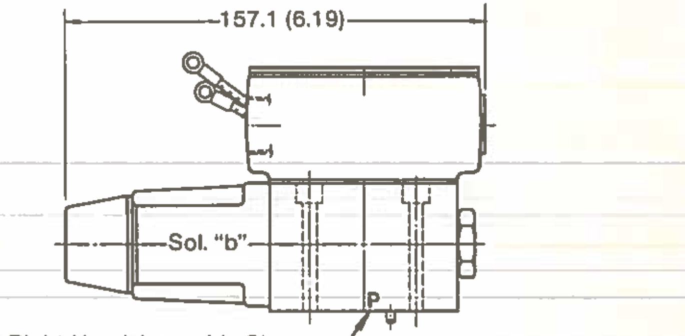
- Note: Mounting Bolt Spacing Helps To Insure Proper Mounting and Port Relationship. Right Hand Single Solenoid Models Have The Solenoid Adjacent To The "A" Port End And The Narrow 31 (1.22) Mounting Bolt Spacing.



KDG4V-3-*CS-M-W-*-40(-S440)**
- Wiring Housing With 1/2 NPT Conduit Connections
- Double Solenoid Models

SINGLE SOLENOID MODELS
KTG4V - 3 - B/F**S - M - W - * - 40

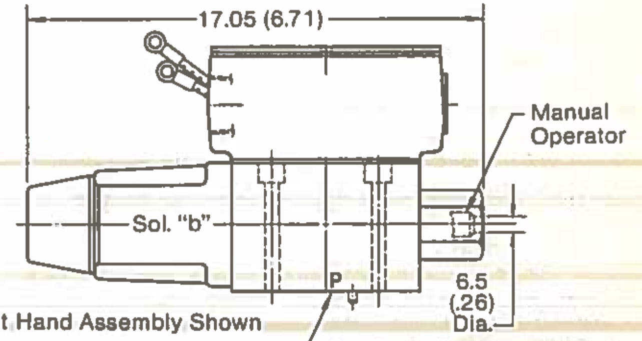
Models:
KTG4V-3-*B**S-M-W-*-40
KTG4V-3-*F**S-M-W-*-40
- Single Solenoid Models Only



Right Hand Assembly Shown

BODY STAMPED TO IDENTIFY PRESSURE PORT SIDE.

Models:
KTG4V-3-*BP**S-M-W-*-40
KTG4V-3-*FP**S-M-W-*-40
- Single Solenoid Models (Only) With Manual Operator In End Cap



Right Hand Assembly Shown

BODY STAMPED TO IDENTIFY PRESSURE PORT SIDE.

Flow Characteristics

Flow ratings are shown in the chart at right. They are stated in liters per minute and (GPM) at 10 bar (145 PSI) pressure drop across the valve. The single solenoid, spring centered KTG4V-3 models are rated to 38 l/min. (10 GPM) through each leg. Therefore, when double porting the KTG4V-3 versions, a maximum flow of 76 l/min. (20 GPM) can be handled.

FLOW RATINGS 4

Valve Type	Model	Flow Capacity ** @ 10 bar (145 PSI) ΔP	Maximum flow @ 350 bar (5000) PSI Litres/Min.	Maximum flow @ 350 bar (5000) PSI U.S. GPM	Prior (-12) Design Cross Reference
Double Solenoid Spring Centered 4	KDG4V-3-2C08	8.0 l/min. (2.1 U.S. GPM)	19	5	-S427
	KDG4V-3-2C15	16.0 l/min. (4.2 U.S. GPM)	38	10	-S428
Single Solenoid Spring Centered Single Ported	KTG4V-3-2B19	24.3 l/min. (6.4 U.S. GPM)	38	10	-S431
Single Solenoid Spring Centered Double Ported		48.4 l/min. (12.8 U.S. GPM)	76	20	

Flow Ratings Litres/Min. (U.S. GPM)

**Flow ratings of the previous -12 design valves were figured @ 200 PSI ΔP. Therefore, the ratings of -40 design at maximum flow are the same.

1. The pressure drop curves give approximate pressure drop (ΔP) when passing 21 cSt (100 SUS) fluid(s) having .865 specific gravity.
2. For any other viscosity(s), cSt (SUS) the pressure drop (ΔP) will change as follows:

Other Viscosity(s)	14 (75)	32 (150)	43 (200)	54 (250)	65 (300)	76 (350)	86 (400)
% of ΔP (Approx.)	93	111	119	126	132	137	141

3. For any other specific gravity (G_1) ★, the pressure drop (ΔP_1) will be approximately: $\Delta P_1 = \Delta P\,(G_1/G)$

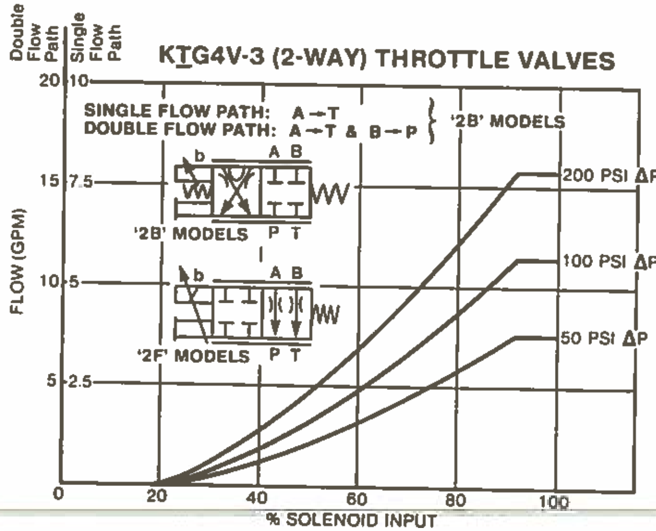
★ Specific gravity of fluid may be obtained from its producer. The value is higher for fire-resistant fluids than for oil.

4 Flow must not exceed the ratings shown. A hydrostat can be used to limit flow in circuits with multiple branches or where pump output flow exceeds the rating of the KDG4V-3 valve. Flow ratings apply to the KDG4V-3-2C***S440 also.

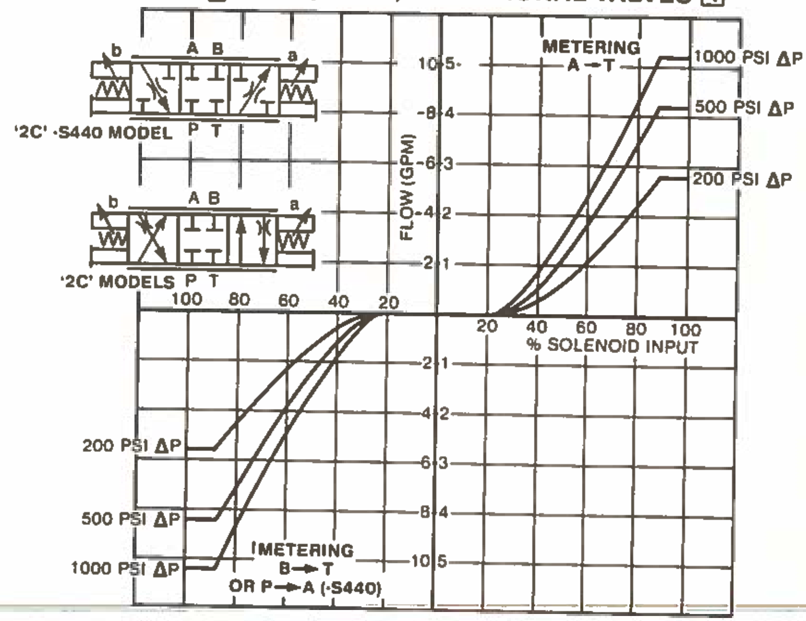
Pressure Drop Curves

Flow output vs. percent of input signal to the valve solenoids is shown as it relates to pressure drop. In the double solenoid models, flow is metered from "A" to "T" or from "B" to "T". There is essentially no metering on "P" to "A" or "P" to "B" flow paths. The -S440 is an exception.

KTG4V-3 (2-WAY) THROTTLE VALVES



KDG4V-3 (4-WAY) DIRECTIONAL VALVES 4



INSTALLATION DATA

2 Mounting Face Information

When mounting to a manifold instead of a subplate, machine the mounting surface as shown within the shaded area of the subplate above. Mounting face must be flat within 0.013mm (.0005) and smooth within 1.1 micrometer (45 microinch). Mounting bolts when provided by customer, should be 12.9 (SAE grade 7), or better.

Subplate and Bolt Kits

Valves, subplates and bolt kits must be ordered separately.
Example: One (1) KDG4V-3-2C15S-M-W-G-40 Valve
One (1) DGVM-3-10-S Subplate
One (1) BKDG-3-698 Bolt Kit

Mounting Bolt Kits

U.S.	BKDG3-698
Metric	BKDG3-699M

Bolt Torque

4.5 to 6.0 Nm (40 to 53 lb.(f)in.)

Adapter Plates

The KDG4V-3 can be mounted in place of the DG4S4-01••-50 series through the use of mounting adapter plate DGAM-3-01-10, shown on drawing 517360. The overall installation height will increase by approximately 38mm. Bolts and seals are furnished for mounting the adapter plate.

Mounting Position

Good hydraulic design practice suggests that detented models can be mounted with longitudinal axis horizontal. Other models may be mounted in any position.

FILTRATION

For satisfactory service life of these components in industrial applications, use full flow filtration to provide fluid which meets ISO cleanliness code 16/13 or cleaner. Selections from Vickers OFP, OFR and OFRS series are recommended. Refer to Section "P"-"Filters" and Section "S"-"Technical Information" of Vickers Catalog 400, or contact your Vickers representative for further filtration advice.

FLUIDS AND SEALS

Viton seals are standard and suitable for use with phosphate ester type fluids or its blends, water glycol, water-in-oil emulsion fluids and petroleum oil. Refer to Vickers data sheet I-286-S, "Hydraulic Fluid and Temperature Recommendations for Industrial Machinery" (included in the Technical Information section of Vickers Catalog 400) for hydraulic fluid and temperature recommendations. High water base fluids are not recommended.

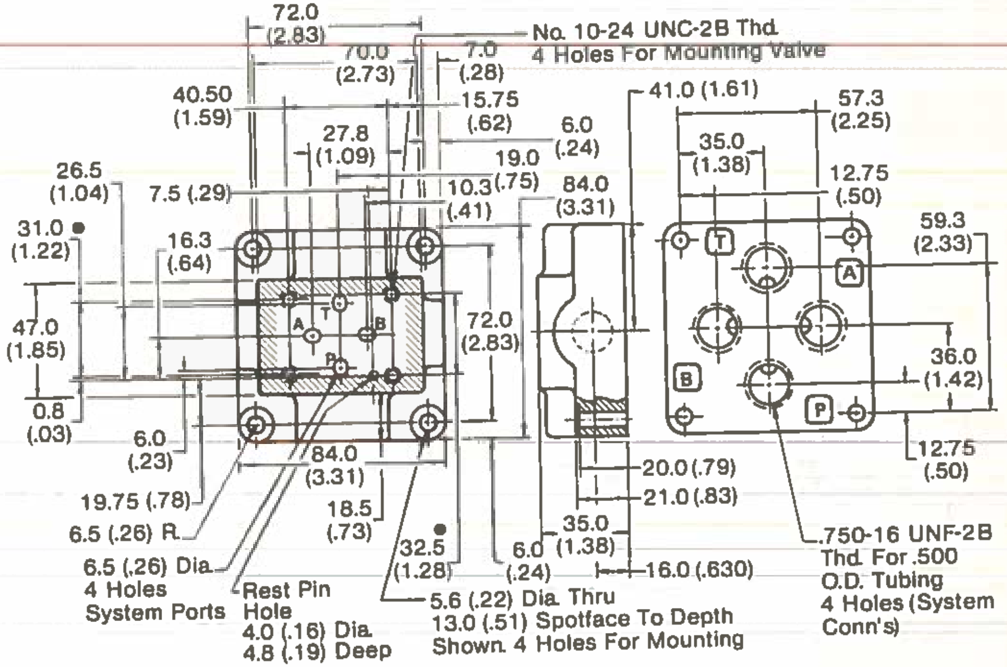
Note:

Any sliding spool valve, if held shifted under pressure for long periods of time, may stick and not spring return due to fluid residue formation and, therefore, should be cycled periodically to prevent this from happening.

WEIGHT

KDG4V-3-•C Double Solenoid Models	2.1 Kg (4.6 lbs.)
KDG4V-3-•B Single Solenoid Models	1.7 Kg (3.7 lbs.)

MOUNTING SUBPLATE
MODEL DGVM-3-10-S



● **Note:** Mounting bolt spacing helps to insure proper mounting and port relationship.

A side port subplate (DGMS-3-1E-10-S) is also available. It is the same size and has the same size ports as the one shown above. Also available is the model DGMS-3-1 EY-10-S with .5625-18 UNF-2B st. thd. ports.

Printed in U.S.A.

ELECTROHYDRAULIC PROPORTIONAL SOLENOID VALVES

MODEL SERIES KDG1 - 3/5A

DIRECTIONAL TYPE

GENERAL DATA

Proportional directional control valves can be used where one or more of the following requirements exist:

- Remote control of machine actuator speeds and direction of movement.
- Speed and directional control in accordance with machine operating cycles or programs.
- Meter-in/meter-out throttle (restrictor) control by the proportional valve itself.
- Pressure compensated flow control for the P to A and P to B meter-in functions, or the A to T meter-out function, with the aid of an appropriate hydrostat module.
- Smooth adjustable control of machine acceleration and/or deceleration.
- Suitability for industrial or mobile applications.

DESIGN AND CONSTRUCTION

The proportional control directional valve is position controlled, i.e. the input signal to the amplifier is continuously compared with the actual value from the valve spool position sensor and any deviations which occur are eliminated by control action. This also overcomes any static and dynamic forces acting on the spool.

RATINGS

Flow	
KDG1-3A	10 or 20 l/min. (2.6 or 5.3 GPM)
KDG1-5A	40 or 80 l/min. (5.3, 10.6, or 21.1 GPM)
Maximum Operating Pressure	315 bar (4500 PSI)
Power Supply	24V dc via p.c.b. amplifier to 12V dc solenoid
Typical Hysteresis	⩽1 percent
Mounting Surface	ISO 4401 sizes 03 and 05 (NFPA-DO1 and DO2, plus drain port)

Travel of the valve spool from center to the required position is proportional to the input signal. The travel is measured by an inductive position sensor and fed back to the electronic amplifier. Any difference between input signal (voltage) and feedback signal (voltage proportional to travel) is calculated and corrected; thus the valve is in an electric position-control circuit. It is unaffected by disturbance forces, responds to the input signals rapidly and precisely and possesses a very low hysteresis.

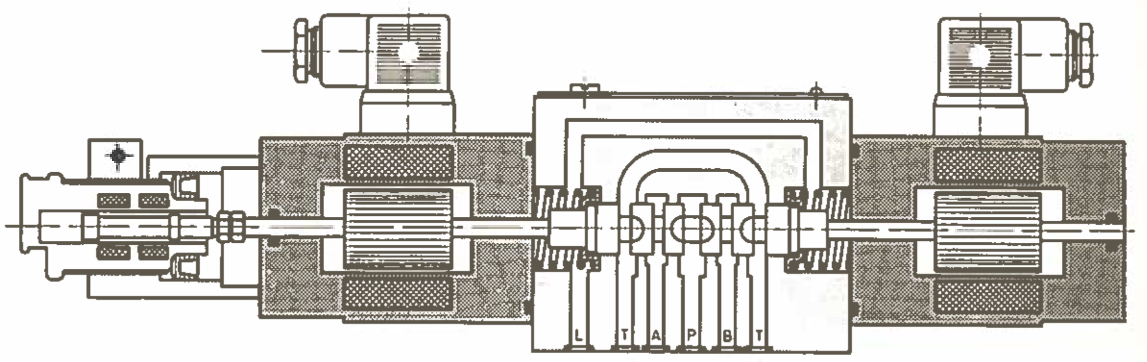
The flow rate is determined by the cross section of aperture and pressure difference across the spool. Fluctuations in pressure difference will cause changes in flow rate but they can be offset by pressure compensation obtained by adding the appropriate pressure hydrostat module under the valve.

ELECTRONICS

These proportional solenoid valves can be used with or without microprocessor controls. Vickers standard matched electronics are available. For full details see installation drawing 521565 (included in the "Electronics" section of Vickers Catalog 400).

Typical Single-Stage Proportional Valve

Proportional directional valve, KDG1-5A-2S



VICKERS®

Vickers, Incorporated
A Libbey-Owens-Ford Company
Troy, MI 48084

INSTALLATION DATA

SINGLE STAGE DIRECTIONAL VALVES

MODEL SERIES
KDG1 - 3A ISO-4401-03 (NFPA-DO1) INTERFACE
KDG1 - 5A ISO-4401-05 (NFPA-DO2) INTERFACE

WITH SPOOL POSITION (ELECTRICAL) FEEDBACK

DWG. NO. 519360

PROPORTIONAL VALVES

K

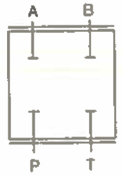
Revised 1-86

APPLICATION GUIDANCE

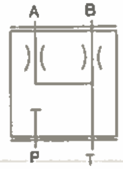
Spool Options

The proportional valve is available with alternative spool center conditions as below.

KDG1-*A-2S Models



KDG1-*A-6S Models



Throttle (Restrictor) Flow Usage

The proportional valve, when used without a hydrostat, provides meter-in or meter-out flow restriction (non-compensated), as appropriate, through all flow paths.

Pressure Compensated Flow Arrangements (P to A and B)

Obtained by using the proportional valve with a KDGMH-3-616261 (Δ p = 8 bar) or KDGMH-5-616266 (Δ p = 5 bar)* hydrostat as appropriate. This type of hydrostat cannot be used with the A-to-T type.

Pressure Compensated Flow Arrangement (A to T)

Obtained in size 3 stacks by using a proportional valve with a KDGMH-3-616263 (Δ p = 8 bar) hydrostat. This type of hydrostat is not currently available for size 5 stacks and neither can it be used with the P-to-A and B type.

Proportional Valves Connected in Parallel

In such cases each proportional valve must be used with its own P to A and B hydrostat. Any parallel-connected valves other than these proportional valves must be fed through suitable hydrostats or pressure reducers.

Drain Option

See maximum operating pressure.

OPERATING DATA

Maximum Operating Pressure

Ports P, A and B; Valves and Hydrostats .... 315 bar (4500 PSI)
Port L ♦; Valves and Hydrostats:
  Size 3 Units .... 160 bar (2300 PSI)
  Size 5 Units .... 2 bar (30 PSI)
Port T; Valves and Hydrostats:
  Size 3 Units (Port L externally drained) .... 250 bar (3600 PSI)
  Size 3 Units (Port L ▲ internally drained to T) .... 160 bar (2300 PSI)
  Size 5 Units ● .... 315 bar (4500 PSI)

▲Port L must be blocked by the mounting subplate or pad hence in this case alternative subplate is DGVM-3-10-S.

♦ For size 3 valves 'L' must be used and connected directly to the reservoir if "T" port pressure exceeds 160 bar (2300 PSI).

● For size 5 valves use of the 'L' drain port is recommended for all applications.

Electrical Data

Solenoid Voltage (Maximum) .... 12V DC
Current @ 50°C (122°F):
  KDG1-3 Models .... 2.7A maximum @ 12V DC
  KDG1-5 Models .... 3.7A maximum @ 12V DC
Power Consumption @ 50°C (122°F):
  KDG1-3 Models .... 32.5 W maximum
  KDG1-5 Models .... 42.5 W maximum
Relative Duty Factory .... 100 percent continuous rating
Type of Protection:
  Winding Insulation to Class F
  Epoxy Resin to Class F
Electrical Connections:
  Solenoids .... DIN 43650 connectors supplied with valve.
  Position Sensor .... Special connector supplied with valve.
Note: Do not use connectors with indicator lights (refer to 519360-8).

MODEL CODE

For Hydraulic Valves

KDG1 - * A - * S - **** - 10**

1 2 3 4 5

1 Proportional Directional Valve

Single stage, with spool position (electrical) feedback.

2 Mounting Size

3 - ISO 4401 (NFPA - DO 1)
5 - ISO 4401 (NFPA - DO 2)

3 Spool Type

2 or 6. See "Spool Options" on 519360-2.

4 Assembly Number

Relative to size, spool, solenoid energization, and rated flow @ Δ p = 8 bar (116 PSI):

KDG1-3A-2S-**614871**	10 l/min. (2.6 GPM)
KDG1-3A-2S-**614872**	20 l/min. (5.3 GPM)
KDG1-3A-6S-**614873**	10 l/min. (2.6 GPM)
KDG1-3A-6S-**614874**	20 l/min. (5.3 GPM)
KDG1-5A-2S-**614876**	40 l/min. (10.6 GPM)
KDG1-5A-2S-**614877**	80 l/min. (21.1 GPM)
KDG1-5A-6S-**614879**	40 l/min. (10.6 GPM)
KDG1-5A-6S-**614880**	80 l/min. (21.1 GPM)

5 Design Number

Subject to change. Outside dimensions remain as shown for numbers 10 through 19.

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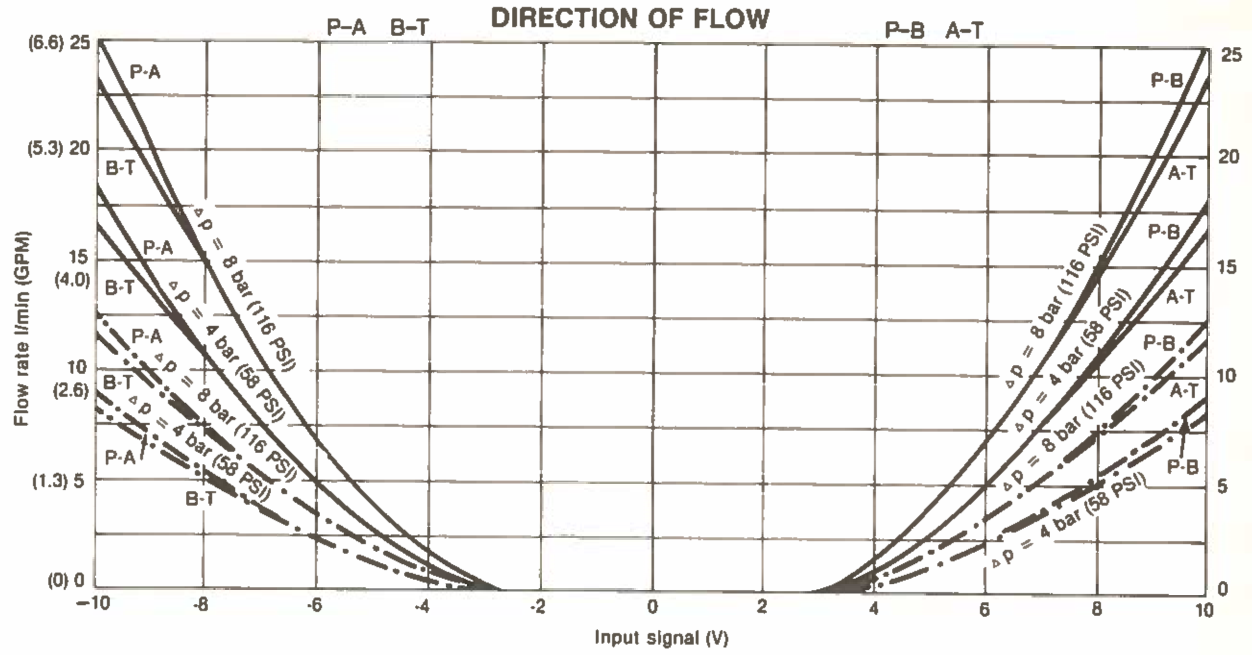
* Alternative hydrostat for higher flows at Δ p = 8 bar, available on request.

KDG1-3A-2S Proportional Valves

— · — · — · — KDG1-3A-2S 614871

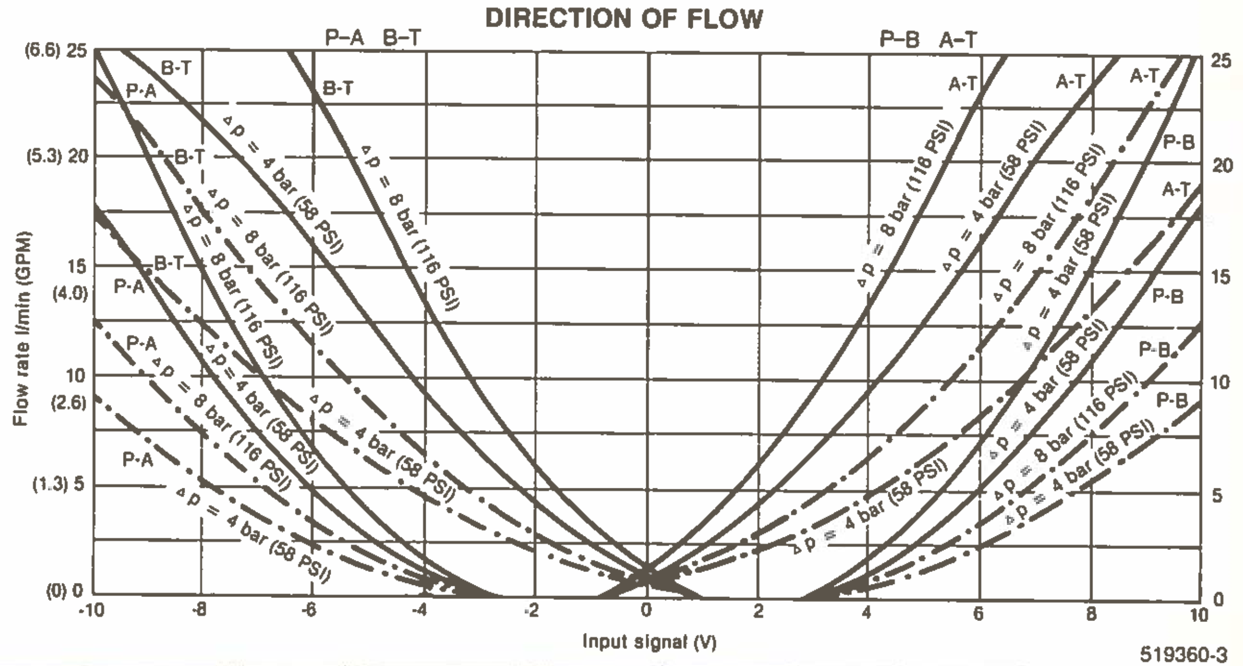
———————— KDG1-3A-2S 614872



KDG1-3A-6A-6S Proportional Valves

— · — · — · — KDG1-3A-6S 614873

———————— KDG1-3A-6S 614874



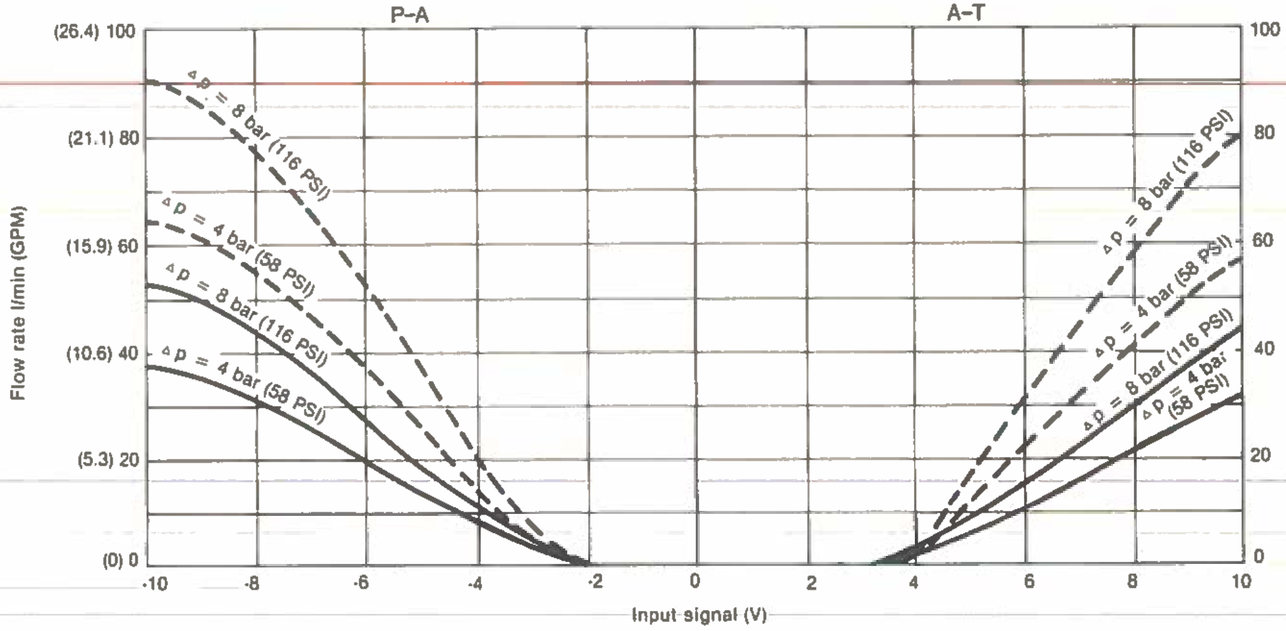
519360-3

KDG1-5A-2S Proportional Valves

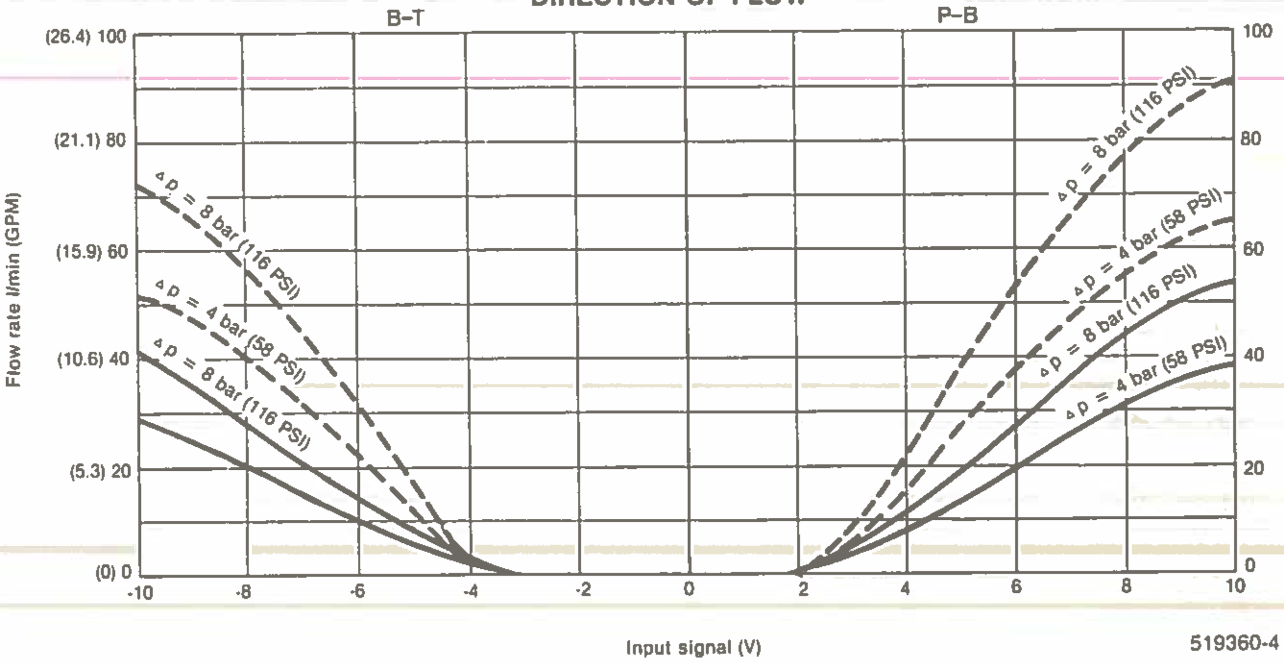
—————— KDG1-5A-2S 614876

– – – – – – KDG1-5A-2S 614877

DIRECTION OF FLOW

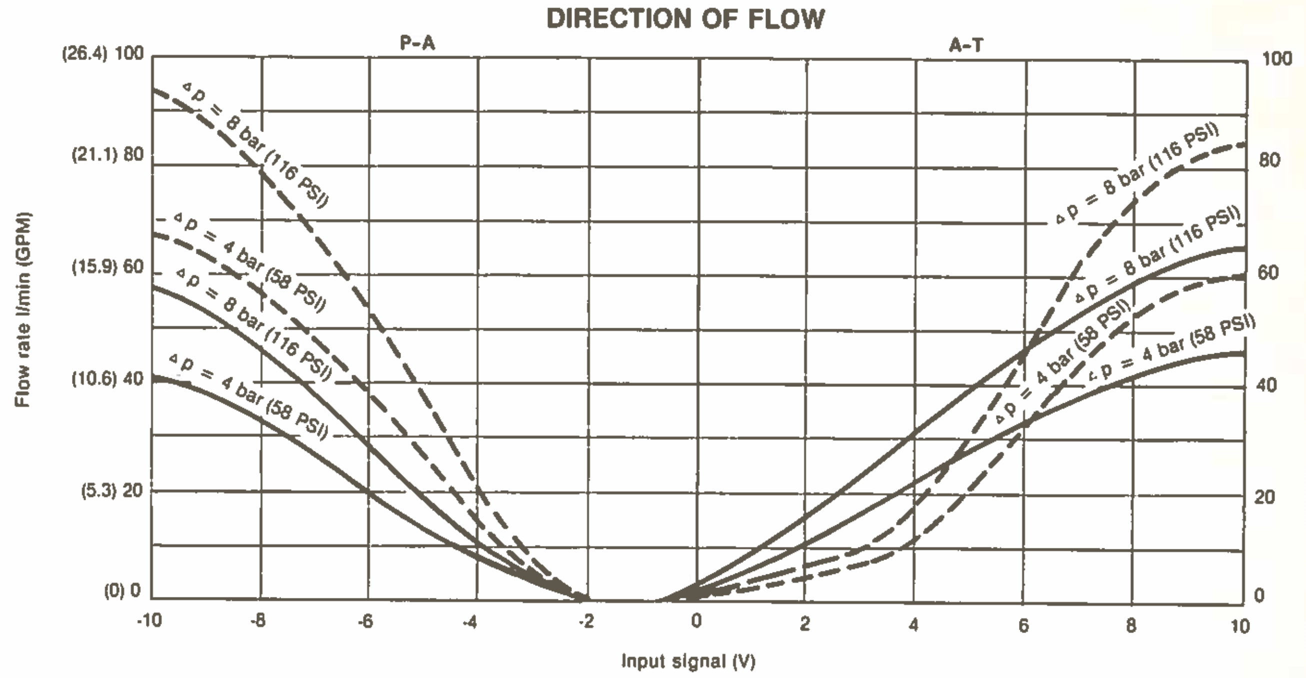


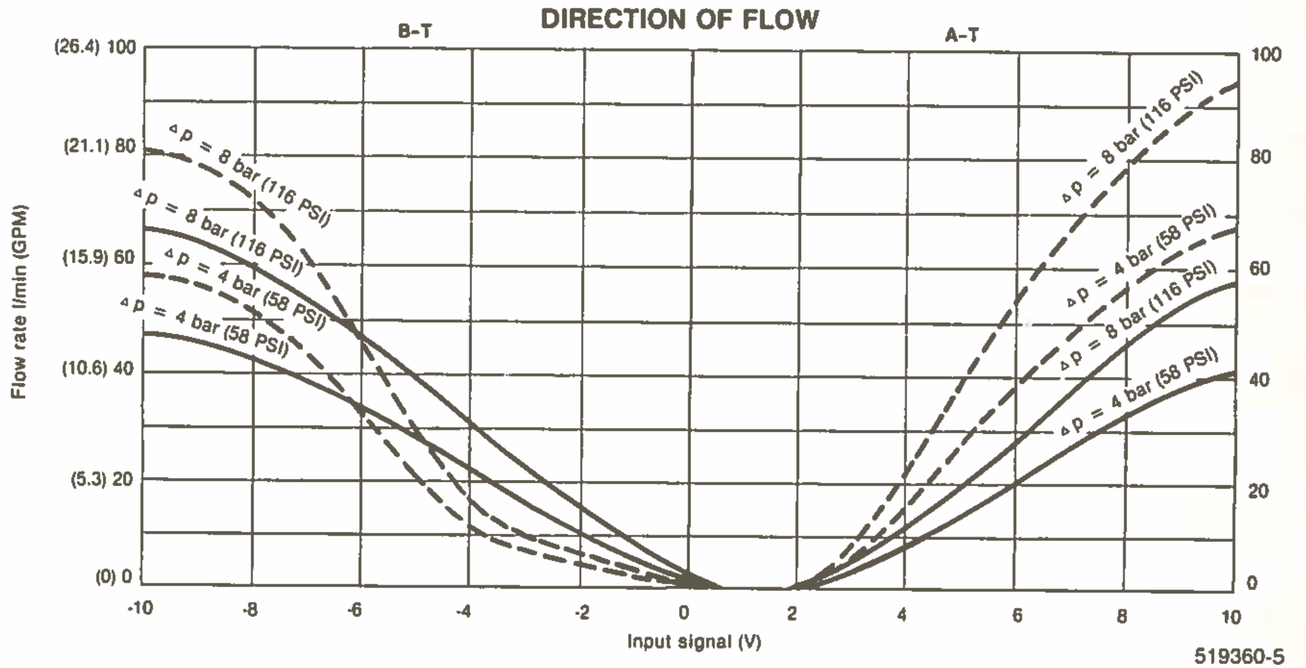
DIRECTION OF FLOW



——————— KDG1-5A-6S 614879

– – – – – – – KDG1-5A-6S 614880


DIRECTION OF FLOW


DIRECTION OF FLOW

Performance Data

Unless otherwise stated, performances are based on mineral oil at 35cSt (160 SSU) and at 50°C (122°F).

Flow Characteristics

For a given input signal value the characteristics of the valve approximate to the theoretical square-law relationship for sharp-edged orifices:

$$\Delta p_x = \Delta p_D \bullet \left(\frac{Q_x}{Q_D}\right)^2$$

Where Δp_x = Pressure drop for required flow rate Q_x

Δp_D = Pressure drop at a datum flow rate Q_D

Q_x = Flow rate for which Δp_x is to be calculated

Q_D = Flow rate at datum Δp_D

Flow Gain

Note: When a KDG1 valve is used with a hydrostat (See "Application Guidance" section) the resulting pressure compensated flow is effectively that related to the Δ p rating of the hydrostat (see 'Model code' for hydrostat modules.)

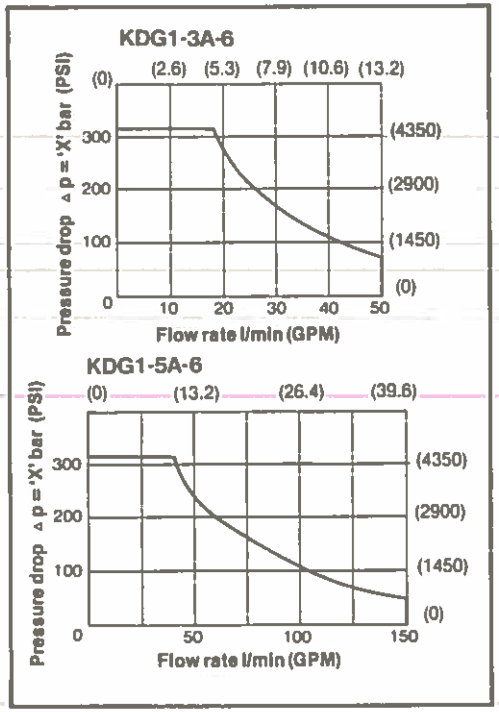
Steady State Dynamic Performance

Hysteresis:
KDG1-3A Valves ≤ 1.0 percent
KDG1-5A Valves ≤ 0.75 percent
Reversal Error:
KDG1-3A Valves ≤ 1.0 percent
KDG1-5A Valves ≤ 0.5 percent
Reproducibility:
KDG1-3A Valves ≤ 0.5 percent
KDG1-5A Valves ≤ 0.3 percent
Response Times:
Approx. for Percentage Spool Travel
KDG1-3A Valves:
10 to 90 percent ...... 40 ms
90 to 10 percent ...... 30 ms
−90 to +90 percent .. 70 ms

KTG1-5A Valves:
10 to 90 percent ...... 55 ms
90 to 10 percent ...... 55 ms
−90 to +90 percent .. 90 ms
Production Tolerance on Flow/ Signal Function .....≤ 5 percent

Maximum Pressure Drop/ Flow Envelopes

KDG1 valves as restrictors non-compensated flow (Looped pressure drop, P to A or B plus B or A to T)



Hydraulic Fluids

All valves recommended for use with antiwear hydraulic oils. The extreme operating viscosity range is from 500 to 10 cSt, but the recommended range is from 54 to 13 cSt.

For further information about hydraulic fluids, see Vickers data sheet I-286-S, "Hydraulic Fluids and Temperature Recommendations for Industrial Machinery" included in the Technical Information section of Vickers Catalog 400. If applications require different types of fluid, consult Vickers, providing details of fluid and operating conditions.

Note:
High water base fluids are not recommended.

Temperature Limits

Oil Temperatures:
Minimum ...... −25°C(−13°F)
Maximum ...... +80°C(176°F)

To obtain optimum service life from both fluid and hydraulic system, 65°C (150°F) is the recommended maximum.

Whatever the actual temperature range, ensure that viscosities stay within the limits specified in the "Hydraulic Fluids" section.

Filtration Requirements

For satisfactory service life of these components in industrial applications, use full flow filtration to provide fluid which meets ISO cleanliness code 16/13 or cleaner. Selections from Vickers OFP, OFR and OFRS series are recommended. Refer to Section "P"-"Filters" and Section "S"-"Technical Information" of Vickers Catalog 400, or contact your Vickers representative for further filtration advice.

WEIGHT

KDG1-3A-*S
Proportional Valve ....6.0 lbs.
KDG1-5A-*S
Proportional Valve ... 18.7 lbs.
KDGMH-3-61626
Hydrostat ....2.4 lbs.
KDGMH-3-616263
Hydrostat ....2.4 lbs.
KDGMH-5-616266
Hydrostat .... 13.7 lbs.
KDGVM-3 Subplate .....3.3 lbs.
KDGSM-5 Subplate .....4.5 lbs.

INSTALLATION DATA

Mounting Attitude

Unrestricted.

Drain Line "L"

See "Maximum Operating Pressures" section.

ORDERING PROCEDURE

The various hydraulic components must be ordered as, and are supplied as, separate items except where otherwise indicated. State full model designations, e.g.:
(2) KDG1-3A-2S-614872-10
proportional valve
(2) KDGMH-3-616261-10
hydrostat
(2) KDGVM-3-676803-10
subplate

The above equipment is likely to be needed where two separate machines require:

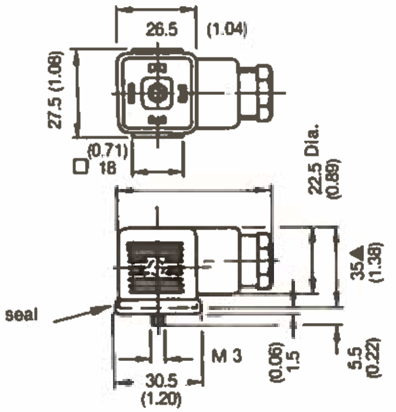
- Bi-directional movements requiring up to 20 l/min. (5.3 GPM) @ 315 bar (4500 PSI).
- Pressure-compensated flow from P to A and B ports; flow from A to T and B to T being non-compensated.

INSTALLATION DIMENSIONS

Plug Connectors (DIN 43650)

For Proportional Valve Solenoids
(supplied with valve):
977897-Grey Plug, marked "A"
(solenoid 'a')
977896-Black Plug, marked "B"
(solenoid 'b')

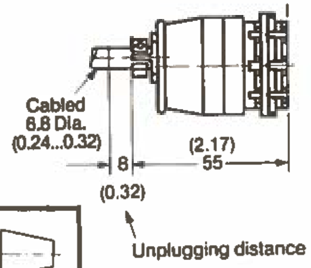
Means of Connection: Screw Terminals
Conductor Cross Sectional Area: .5 to 1.5mm²
Cable Diameter: 6 to 10mm



▲ May vary depending on source.

Sensor Cable Connector
(supplied with valve)

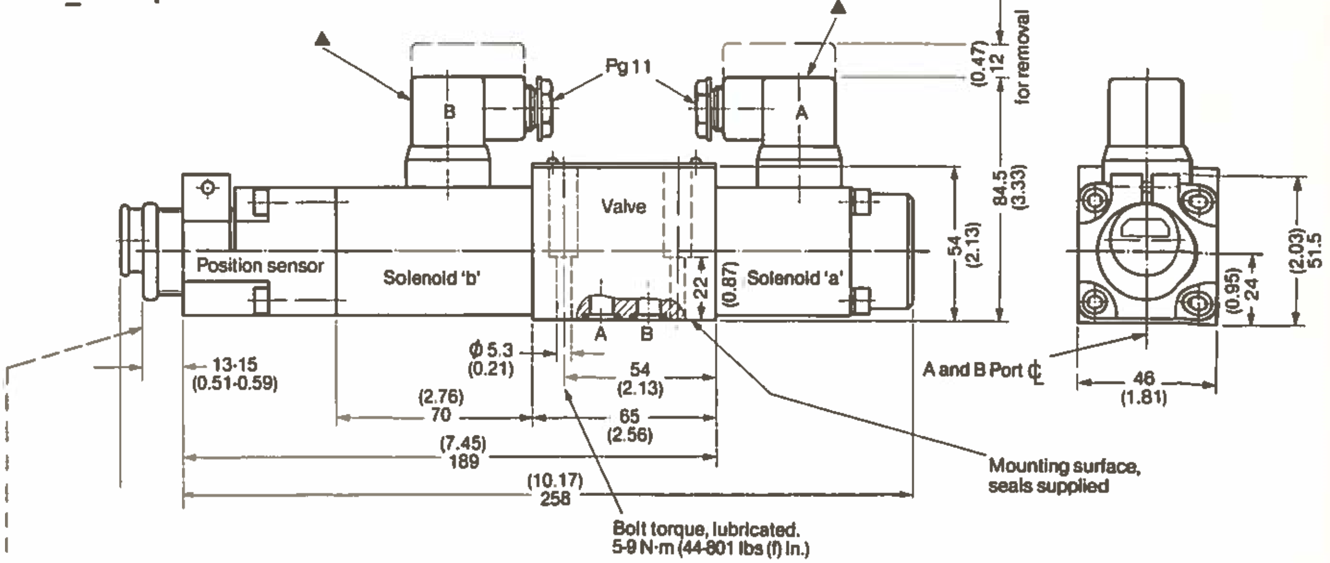
Part No. 709888



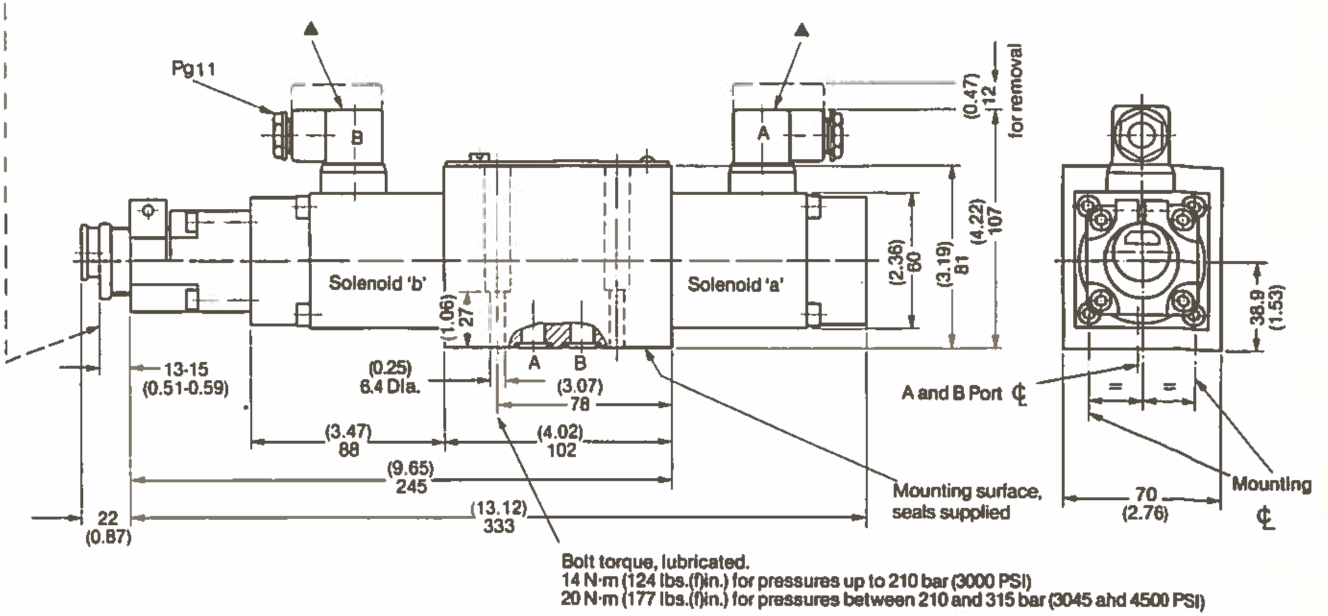
1st Angle Projection

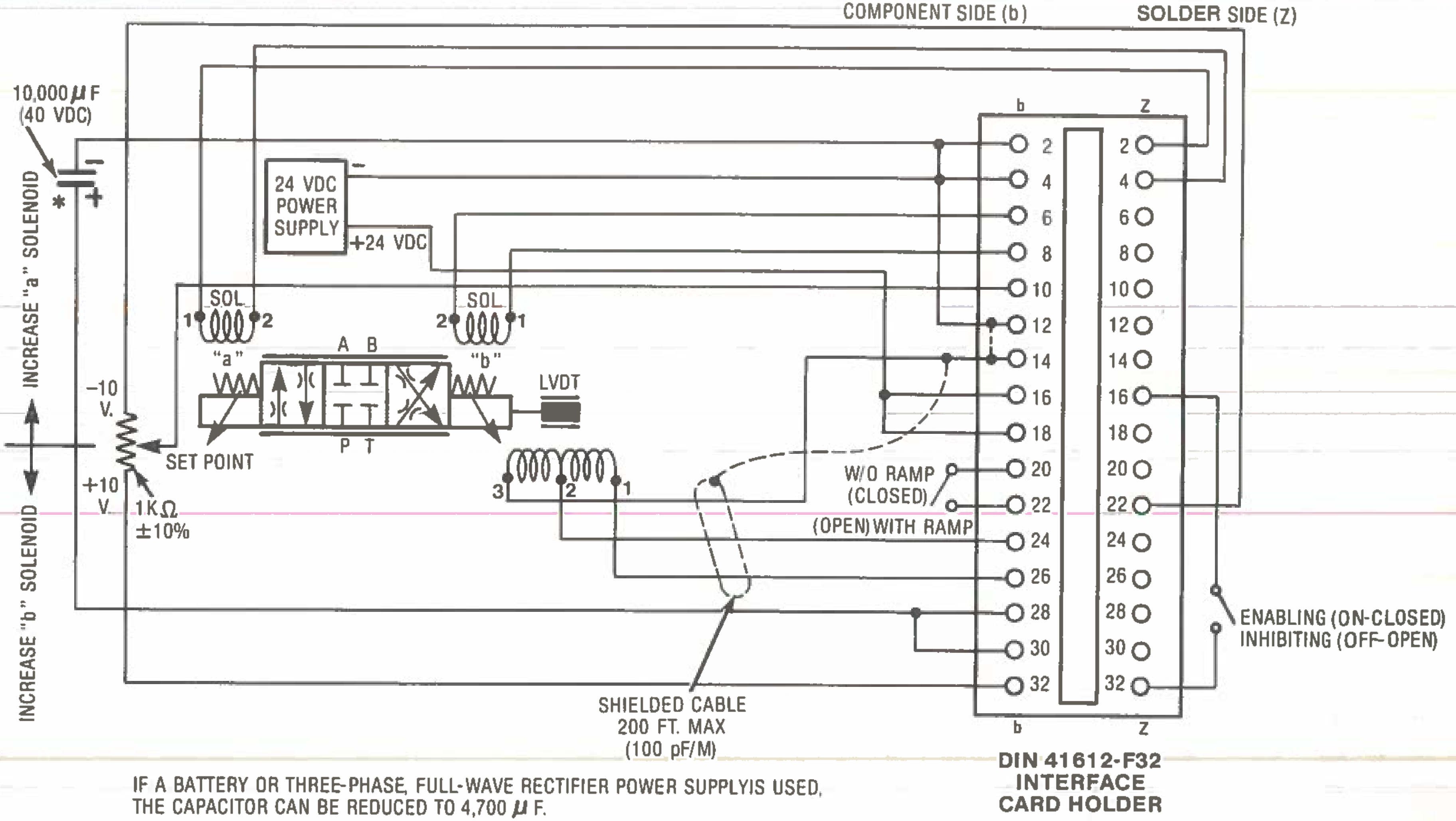
DIMENSIONS SHOW IN MILLIMETERS AND (INCHES)

KDG1-3A Proportional Valves



KDG1-5A Proportional Valves



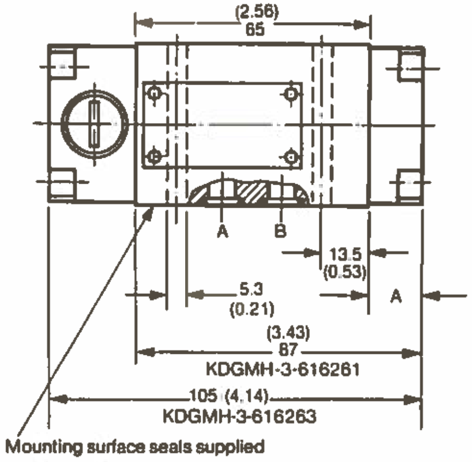


IF A BATTERY OR THREE-PHASE, FULL-WAVE RECTIFIER POWER SUPPLYIS USED, THE CAPACITOR CAN BE REDUCED TO 4,700 µ F.

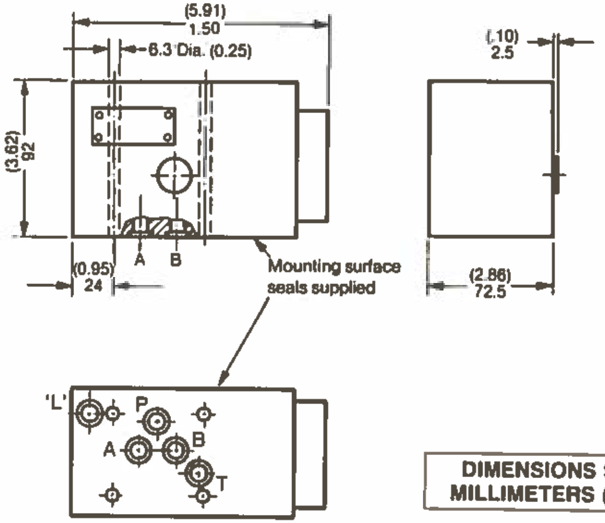
IF THE SUPPLY IS TAKEN FROM A SINGLE PHASE FULL-WAVE RECTIFIER, IT IS ESSENTIAL THAT THE CAPACITOR BE USED.

HYDROSTAT MODULES

KDGMH-3-616261
KDGMH-3-616263

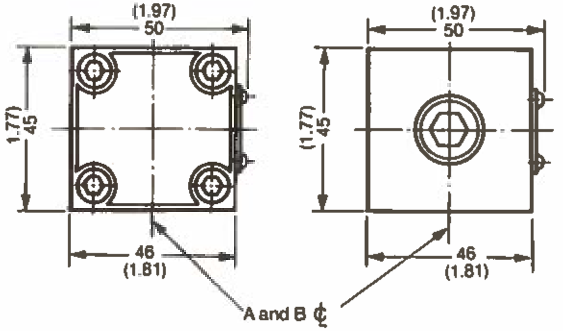


KDGMH-5-616266

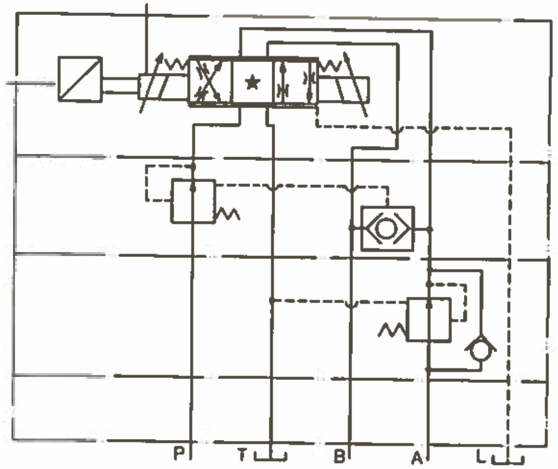


DIMENSIONS SHOWN MILLIMETERS (INCHES)

KDGMH-3-616263 **KDGMH-3-616261**



Model	Dimension "A"
KDGMH-3-616261	22 (.80)
KDGMH-3-616263	15 (.59)



Proportional directional valve

Hydrostat module, P to A & B
KDGMH-3-616261
KDGMH-5-616266

Hydrostat module, A to T
KDGMH-3-616263

Subplate
KDGVM-3-676803-10
KDGSM-5-676805-10
(refer to page 519360-9)

"See 'Application notes'

MODEL CODE
For Hydrostat Modules

KDGMH - * - **** - 10**

1 Hydrostat Module

2 Mounting Size

3 - ISO 4401-03(NFPA - DO 1)
5 - ISO 4401-05(NFPA - DO 2)

3 Assembly Number

Relative to size, function, and Δp rating.
KDGMH-3-616261 . . .P to A and B control, Δp = 8 bar (116 PSI); maximum flow 25 l/min. (6.6 GPM).
KDGMH-3-616263 . . . . . . A to T control, Δp = 8 bar (116 PSI); maximum flow 25 l/min. (6.6 GPM).
KDGMH-5-616266 . . .P to A and B control, Δp = 5 bar (75 PSI); maximum flow 50 l/min. (13.2 GPM).

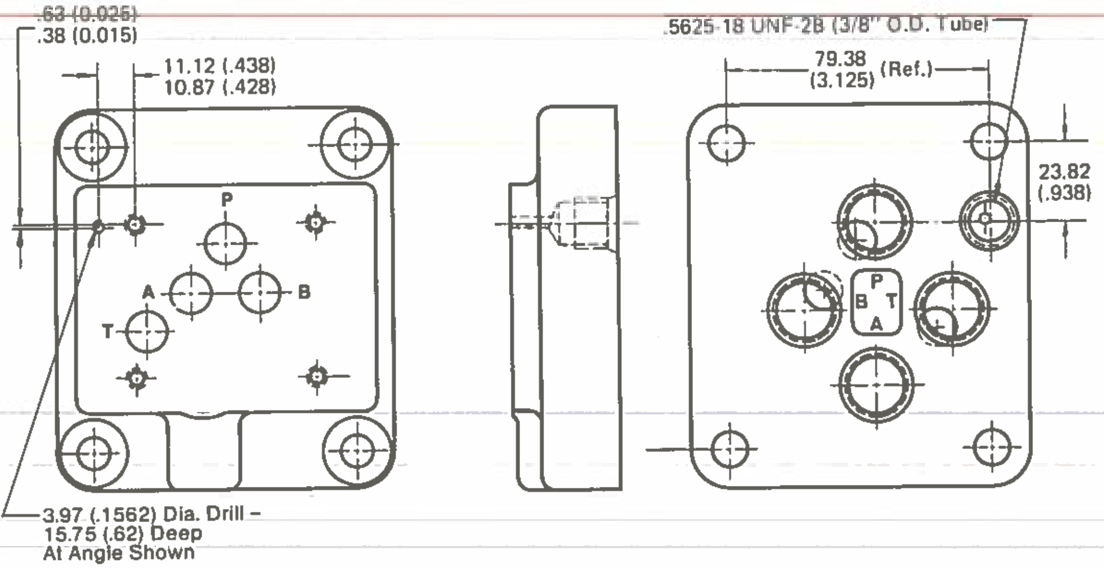
4 Design Number

Subject to change. Outside dimensions remain as shown for numbers 10 through 19.

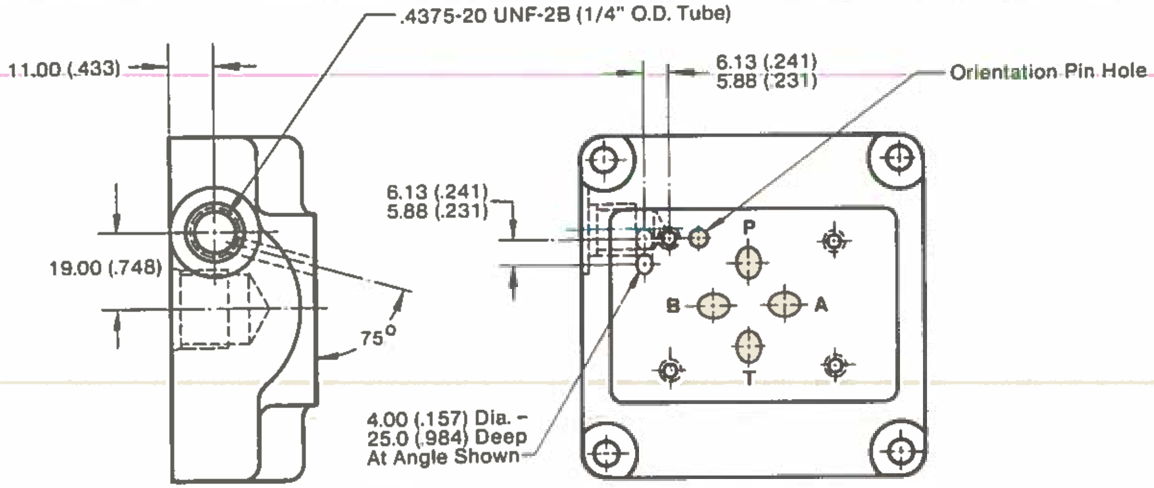
MOUNTING SUBPLATES WITH "L" DRAIN PORT

DIMENSIONS SHOWN MILLIMETERS (INCHES)

MODEL KDGSM-5-676805-20
ASSEMBLY 676805
(MAKE FROM DGSM-01-20-T8)
MAX. PRESSURE 3000 PSI



MODEL KDGVM-3-676803-10
ASSEMBLY 676803
(MAKE FROM DGVM-3-10-S)
MAX. PRESSURE 5000 PSI



For Mounting Bolt
For mounting proportional valves directly to subplates:

Size 3 Valves . . .(4) bolt 10358 (10-24UNC-2B)
Size 5 Valves . . .BKDG-01-633

For mounting proportional valves and modules in stacks (bolts supplied by customer), consult your Vickers representative.

When subplate is not used, a machined pad must be provided for mounting; pad must be flat within .0005 inch and smooth within 63 microinch. Mounting bolts, when provided by customer, should be SAE grade 7, or better.

PVD - OPEN LOOP PUMPS

PVD open loop pumps are single side of center, variable volume piston pumps of the inline design. The basic package includes a full flow charge pump, charge relief valve, a single side of center piston pump, and controls.

The PVD pump is essentially a two-stage pump. The first stage is a gear pump whose displacement is approximately 15% greater than the maximum displacement of the second stage which is a piston pump. A full flow filter is recommended in the external line connecting the gear pump discharge port to the piston pump inlet port. The PVD schematic showing the basic supercharge circuit is on page 10. The gear pump takes fluid directly from the reservoir and in conjunction with the charge relief valve and filter, supplies clean fluid to the inlet of the piston pump. The excess flow from the charge pump is returned to the reservoir. The gear pump was chosen for the first stage because of its excellent fill characteristics and its high resistance to contamination at low operating pressures. It is also a relatively inexpensive service item that serves as the first line of defense for the more expensive second stage.

It should be noted that the PVD charge pump is capable of operating at an inlet vacuum of 5" Hg.

As shown on the schematic on page 10, the full flow filter is located apart from the pump. This is to enable installation in an easily serviced position on the machine. The section on filtration on page 7 contains the specifications for the filters. Precautionary measures should be taken to assure that the maximum clean element pressure drop from the charge pump outlet to the piston pump inlet (Port A to Port B) does not exceed 100 psi. Please note that the filter should be equipped with a by-pass valve. The piston pump will tolerate normal levels of contamination while the filter is by-passing for short periods of time. Cavitation damage would surely occur, however, if a dirty filter blocked all oil flow into the piston pump.

To ensure adequate case cooling, a flow of 1.5 to 2 GPM is circulated from the charge circuit into the piston pump case through fixed orifices in the charge relief valve.

This cooling oil plus normal piston pump and control circuit leakage is routed from the pump case back to tank via the case drain.

The case-to-inlet flapper type relief valve mounted in the valve block prevents transient pressure overshoots inside the case during rapid off-stroking of the piston unit. Without this relief valve, these rapid yoke movements could generate high case pressure transients and cause seal and shoe wear problems. The pump destrokes when the large diameter control piston moves out of its bore and the small diameter bias piston moves into its bore. The net effect is a displacement of oil within the case which must be allowed to escape. External draining of the case is recommended as a "sound hydraulic practice".

Benefits of Full Flow Supercharge

The benefits of full flow supercharge are summarized on page 5. Problems associated with cavitation are eliminated because the piston pump inlet oil is pressurized. A key feature of the PVD pump is that it is capable of higher speed operation than any competetive open loop pump. In many cases this high speed capability allows a smaller displacement pump to be used, thus providing the customer with a cost savings over using a larger pump. Another feature of the full flow supercharge arrangement is the fact that the filter ensures that the piston pump always gets clean fluid. These two features individually and together mean lower cost of operation through greater reliability and less down-time.

Other brands of piston pumps that claim relatively high speed capability may not experience cavitation during steady state operations but when they shift from low to high displacement cavitation could easily occur. Thats because the inlet oil must accelerate while the pump moves to the higher displacement and, without a supercharge pump to help it, only 14.7 psi is available to accomplish this acceleration.

During rapid on-stroking, pump pistons must accelerate out of their bores at rates up to 130 g's if the shoes are to remain in contact with the swash plate. In non-supercharged, inline type units, the pistons are moved out of their bores by mechanically pulling or pushing on the shoes. This creates a stress on the shoes and will result in wear. A more critical area of concern is the piston/shoe ball and socket joint. When the shoe is pulling the piston out of the bore, this socket joint is in tension. During pumping, the joint is in compression.

This severe thrust reversal occures once each revolution and, coupled with relatively dirty oil, makes the piston/shoe socket joints a weak link in competetive inline type pumps. The supercharge pressure of the PVD provides a hydraulic force to assist in accelerating the pistons and decreases the differential of thrust reversal on the critical socket joints, thus reducing the wear on the shoes.

The 1.5 to 2 gpm case cooling flow also contributes to better reliability. Tapered roller bearing life is a function of many factors including operating temperature. By providing a continuous supply of cool oil to the case, we minimize bearing temperatures, thus improving their life and their reliability.

Rotating Group

The rotating groups are of the inline design and contain nine working pistons. The nine equally spaced steel pistons run in a bronze sleeved steel cylinder block. Sintered iron shoes are swaged onto the pistons. The shoes run on a cast iron swash plate which has been hardened using the "Tuftride" process. The cylinder block runs against a bronze coated steel valve plate.

The shoes are positioned under a slotted steel, fixed clearance type shoe retainer. The retainer is used to keep the shoes from tipping up on edge due to centrifugal force. This situation is called shoe roll and results in damage to the outer corner of the shoes which can destroy the hydrostatic balance which is critical to reliable operation. Scoring of the swash plate can also occur during shoe roll.

Another sound maintenence feature of the rotating group construction is its ability to be serviced in kit form similar to the cartridge kit concept used in Vickers VHO and VQ vane pumps.

Housing

The cast iron housing provides rigid support and accurate location for the yoke, valve block, and rotating group assemblies. Special consideration was given to the housings to assure that they will safely support the inherent loads at high pressures. The housing has two 180° opposed SAE straight thread ports incorporating the appropriate plugs. The port having the highest elevation is to be used to fill the case prior to start up and serves as the case drain outlet.

Valve Block

The ductile iron valve block serves to connect the high and low pressure ports to the appropriate ports in the valve plate. Contained within the valve block are the supercharge relief valve and the case pressure relief valve. The control modules and charge pump are mounted on the exterior of the valve block. The bias piston stem and control piston cylinder are screwed into the valve block.

Yoke and Control Pistons

The ductile iron yoke is an integral pintle design which pivots in tapered roller bearings which are mounted in the housing. Its angle is controlled by means of the bias and control pistons which act on bearing pads on the yoke. The maximum yoke angle in the 25 and 35 size units is 18°. It is limited to 16° in all larger size units. The minimum yoke angle on all units is 0°.

Shaft and Seal

The hardened steel main shaft is supported by heavy duty tapered roller bearings. It drives the rotating group by means of a crowned spline. It is possible to order a number of shaft-end configurations but the standard spline should be strongly recommended to minimize parts proliferation and insure satisfactory life when operating at maximum power conditions.

Hefty shaft diameters have been incorporated in the PVD. For example, the PVD45 could transmit an additional 130 horsepower into an auxiliary pump. Stiff shafts also mean less shaft deflection. This is of value because it helps maintain the cylinder block perpendicular to the valve plate. It also contributes to greater bearing life.

The pumps are available with either one or two shaft seals. A single lip type seal is normally used and will provide satisfactory sealing because of the case pressure relief valve which limits internal case pressure.

Where two seals are used the purpose of the outboard seal is to prevent gear box oil from entering the PVD. The inboard seal prevents the hydraulic fluid from entering the gear box. A drain between the two seals can be used to direct leakage to some convenient monitoring point. This is particularly desirable in multiple pump installations where it can otherwise be difficult to determine which pump has a leaking shaft seal.

The shaft seals are held in a retainer which can be removed for seal replacement without additional disassembly of the pump.

Charge Pump

The full flow charge pump is an external gear type which is driven from the main pump shaft with a splined coupling. As stated earlier, its function is to ensure an adequate supply of oil to the piston unit plus a small additional amount for case cooling. The charge pump displacement is approximately 115% of the piston pump displacement.

The charge pumps are essentially modified versions of the standard heavy duty "G" series pumps. Since charge pressure is low, the bearing loads are relatively low and we are able to reduce the length of the package by using shorter journal bearings. The low pressures also enable us to reduct costs by eliminating the flexible side plates and seal packs. A fixed clearance between gears and bodies is established to provide adequate lubrication, field serviceability, and dependable, long, life.

The minimum allowable operating speed for PVD pumps is 500 RPM, but is the charge pump which is the limiting factor. Regardless of speed, the PVD rotating group requires an amount of oil equal to its displacement. Additional oil is required to make up for leakage. The volumetric efficiency of the charge pump decreases with speed, and at 500 RPM, it is able to supply the needs of the piston pump. Larger displacement charge pumps could have been selected to enable lower speed operation, but this was not felt to be necessary since engines generally do not idle at speeds below 500 RPM.

Super Charge Relief Valve

The supercharge pressure is controlled by a relief valve built into the pump valve block.

In the current design, the charge relief valve is a pilot operated sliding spool design. This approach was chosen to give the fast response and low pressure override characteristic required for this application. Because of its fast response, case pressure transients during rapid piston pump off-stroking are minimized. This helps to ensure long charge pump life and permits the use of interstage filters of lower pressure capability. Typical charge relief pressures in the current design are:

Pump	Charge Pressure Range
PVD45	180-205 PSI
PVD80	160-180 PSI
PVD120	250 PSI

The low charge pressure relief valves will maintain a charge pressure of about 55 PSI (PVD45). This is an option and might be encountered when servicing these units.

The low charge pressure relief valve is a single stage, proportional, design. A benefit of this relief valve is reduced power consumption. While the proportional design may allow for a higher percentage of overshoot, this is of little significance since the nominal charge pressure is low. The low charge pressure relief valve cracking pressure is set by an adjusting screw, while the current design valve is adjusted at the factory with shims placed in the pilot valve spring chamber. Consult the appropriate overhaul manual for further details.

Bearings

The shaft and pintle bearings used on the PVD pump are of the heavy duty tapered roller bearing type which provide high load carrying capabilities, long life, and reduced sound levels.

Ports

All ports on the PVD pump are of the SAE type. All are 4 bolt flange ports which conform to SAEJ518C. The piston pump outlet ports are of the high pressure series (6000 PSI). The charge pump ports and piston pump inlet ports are of the standard pressure series.

Mounting Flanges

The PVD pump mounting flanges conform to SAE standard J 744 C.

Fluids

The PVD series is currently qualified for petroleum based fluids. Refer to Parts and Service Literature M-2950-S, which is in the Contamination Section of this workbook.

Filtration

Interstage filtration, as mentioned before, is highly recommended. The preferred filter to use is the OFR or OFRM with 10 micron element and 25 PSI by-pass valve. Additional 10 micron nominal or better return line filtration and a charge pump inlet strainer are base line recommendations which constitute "good hydraulic practices".

PRODUCTION AVAILABLE PVD'S

	PVD 45*	PVD 80*	PVD 120*
Displacement (CIR)	5.77	10.28	15.40
Rated Pressure (PSI)	5000	5000	5000
Minimum Speed	500	500	500
Rated Speed	2700	2400	2100
Rated Flow (GPM)	68	107	140
Hydraulic H.P.	200	310	408

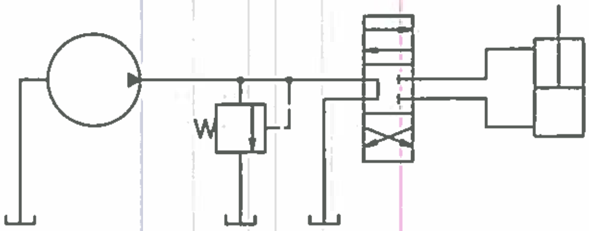
*Theoretical flow at 1800 RPM

Pumps are permitted to operate at maximum speed of 110% of rated speed under the following unloaded conditions:

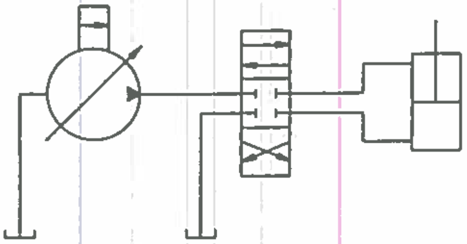
1) Load sensing stand-by.
2) No demand flow in open-center systems.
3) At cut-off on pressure limiter controls.

OPEN-LOOP AND CLOSED LOOP SYSTEMS

The systems pictured in the figures below are known as open-loop systems. The pump receives fluid from a reservoir and the fluid returning from the load or by-passed by the open-center directional valve is returned to the reservoir.



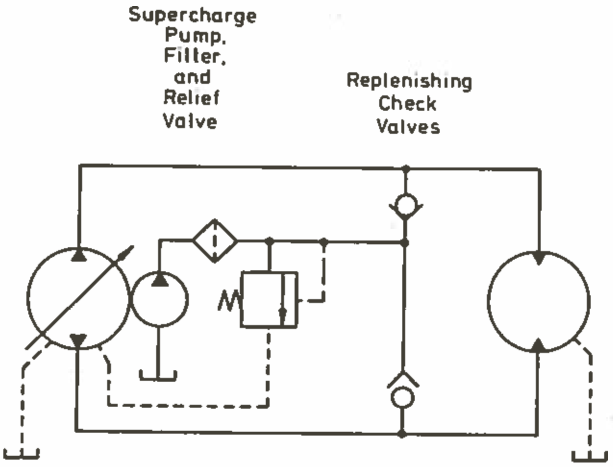
Constant Flow System with Open-Center Directional Valve



Constant Pressure System with Closed-Center Directional Valve

In contrast, a closed-loop is one in which the oil returning from the load goes directly to the pump inlet. The closed hydraulic loop is normally kept filled by a separate supercharge pump. A typical system is shown on page 9. The load is generally a motor capable of continuous rotation.

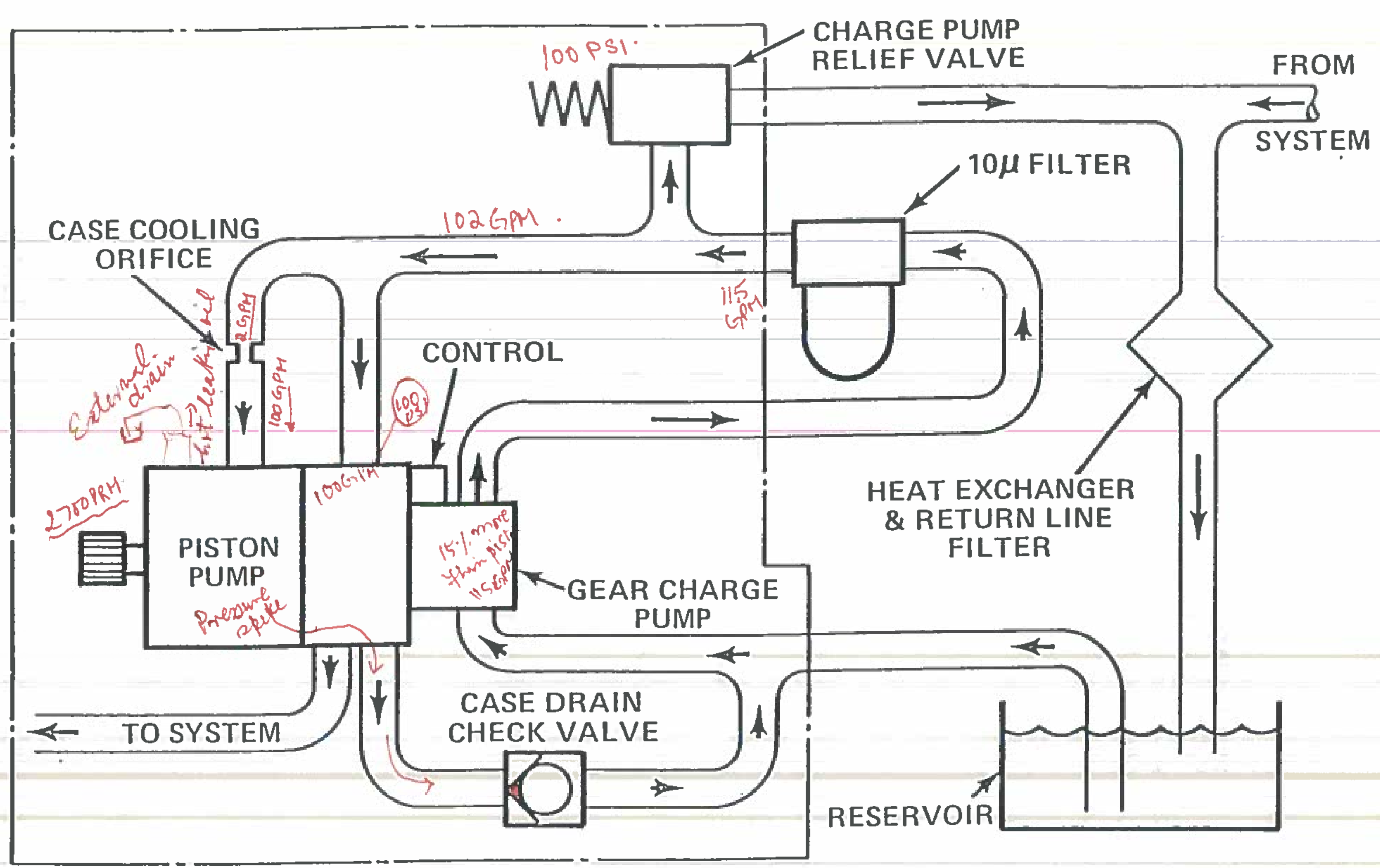
The pump is normally a variable displacement unit, to permit variation in the speed of the motor, and is usually capable of going "over center" to change the direction of flow and reverse the direction of motor rotation. This type of closed-loop system will be discussed more fully in the session covering transmissions.

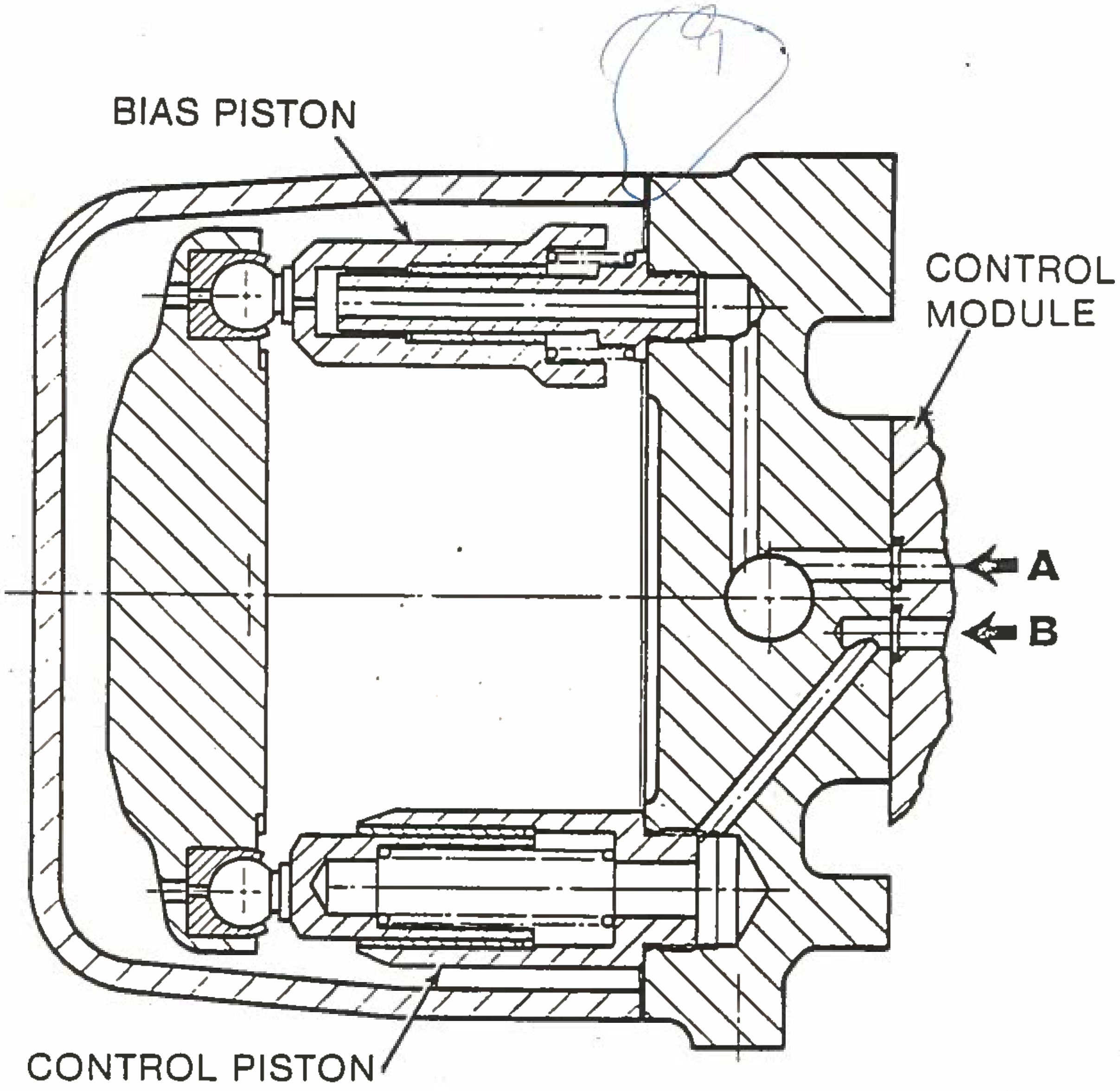

Supercharge Pump, Filter, and Relief Valve
Replenishing Check Valves

Closed-Loop System

PVD SCHEMATIC

Supercharge P.P. (shift the pump progfot)





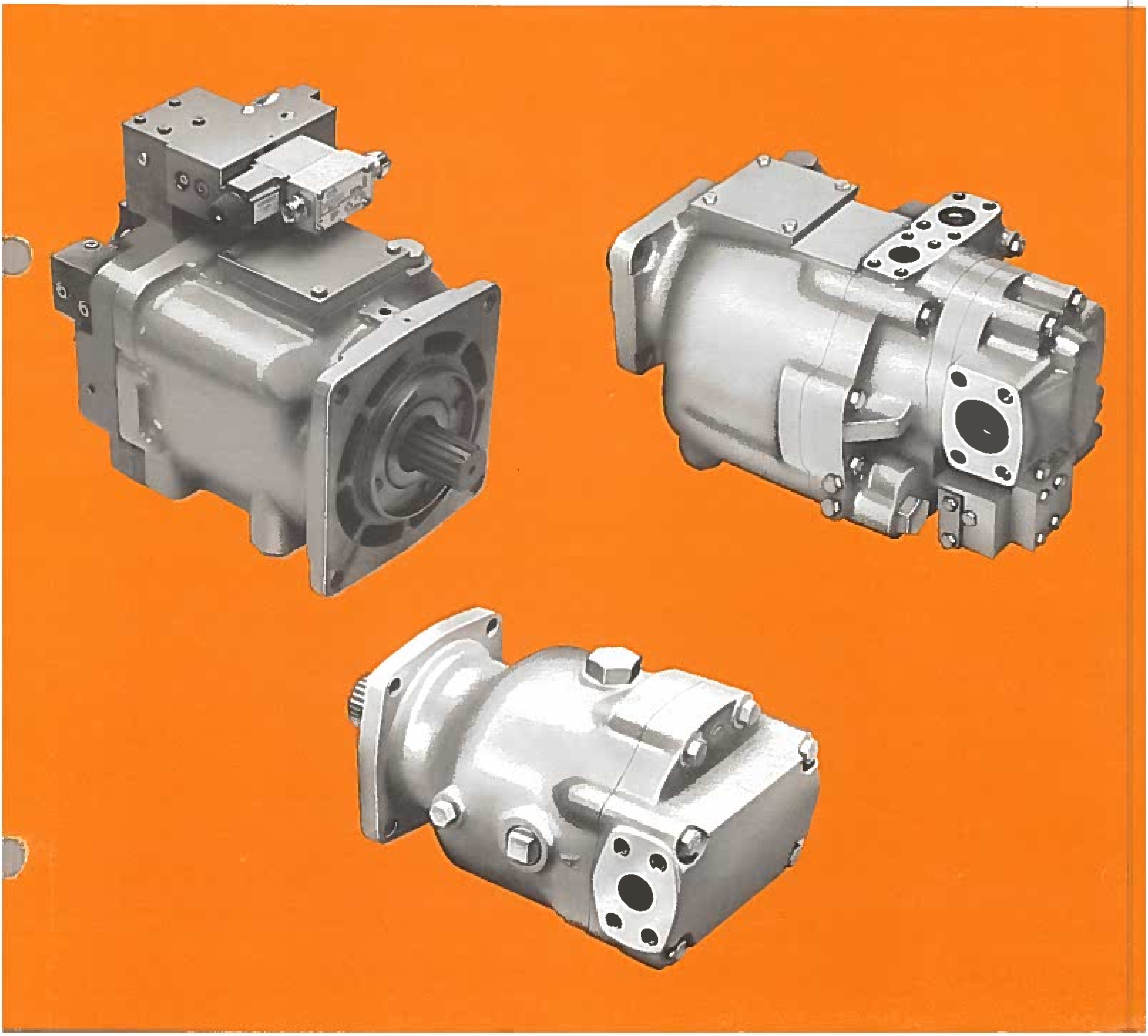
A - PUMP OUTLET PRESSURE
B - CONTROL PRESSURE

PVD PUMP CONTROL PISTONS

⑩

VICKERS®

D Series High Pressure Piston Pumps and Motors for mobile equipment, including POWER MATCH™ Systems



General Description

Pumps

Vickers POWER MATCH heavy duty, variable displacement pumps are for use in open loop, high pressure hydraulic systems. Each unit consists of an inline piston pump with an integral charge relief valve, a full-flow gear type charge pump with an integral case relief valve, and modular pump controls.

Motors

Heavy duty POWER MATCH motors provide a complete hydraulic drive system when paired with POWER MATCH pumps. All motors are available as fixed displacement units; variable displacement units are available in the larger sizes. Motors are available as side ported, end ported, and with provision for direct mounting of POWER MATCH directional valves.

The variable displacement version is a two-position motor for applications which would normally require a two-speed gear box to cover the full torque speed range. The two-position motor gives dual speed capability. Maximum torque, at maximum displacement, is available at speeds up to the rated speed. Higher speeds are achieved at minimum displacement, but at reduced output torques. Various minimum displacements can be made available. Selection of minimum or maximum displacement is accomplished by actuating the motor's hydraulic pilot operated or solenoid control valve.

POWER MATCH
High Pressure Systems

These pumps and motors are designed for use with Vickers pressure compensated, load sensing POWER MATCH directional valves. Together, these units comprise the heart of Vickers POWER MATCH high pressure hydraulic systems. As its name implies, this new system provides a power match between the pump and load, thereby maximizing system and fuel efficiency plus increasing equipment productivity.

Features/Benefits

Reliable, Compact Design

Rugged construction and a 5000 psi (350 bar) continuous duty pressure rating result in a high degree of reliability and a small unit size per horsepower transmitted.

Excellent Bearing Life

Shafts are designed for low stresses and deflection at high loads to maintain optimum alignment and lubricating oil film area with the bearings, thus providing long bearing life.

Heavy Duty Design

The yoke is of an integral pintle design capable of supporting high loads due to system pressure. The valve block has thick sections for minimum deflections and high strength. These components, combined with a large shaft and high-pressure rotating group design, result in extremely rugged units which measure up to the toughest demands of today's construction machinery.

Integral Charge Pump

A full-flow gear type charge pump is mounted directly to the piston pump to provide high speed capability without pressurized reservoirs, and to eliminate cavitation during rapid pump on-stroking. Use of the charge pump also allows a full-flow filter to be ideally located between the charge pump outlet and piston pump inlet for the filtration of system fluid whenever the pump is running.

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Highly Responsive Controls

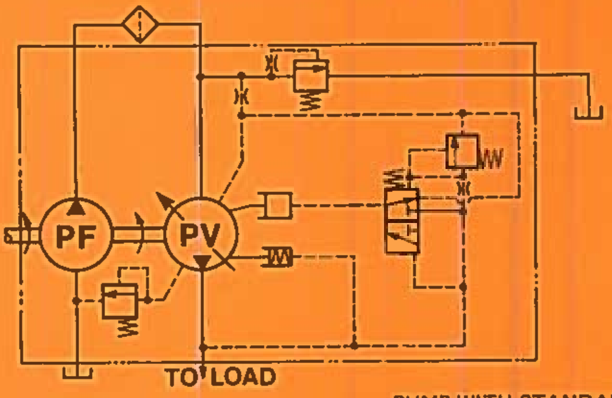
Pressure compensator controls minimize and in many cases eliminate pressure overshoots. This results in longer life for components — hoses, fittings, seals — and prevents high oil temperatures and aeration associated with relief valve flow. Other control options prevent engine stall, match pump output to load requirements, and result in more efficient systems.

Pump Controls

Standard Pressure Compensator (Limiter)

This internally pilot operated control limits system pressure to a desired level and is a standard feature on all pumps. When system pressure reaches the compensator setting, pump displacement starts to decrease to match the load requirement.

The compensator is sensitive to the rate of pressure rise in the system and minimizes pressure overshoot. In a highly responsive system, the pump will begin de-stroking before system pressure reaches the compensator setting. As a result, system shocks which shorten component life and cause leakage are greatly reduced, and in many cases eliminated.

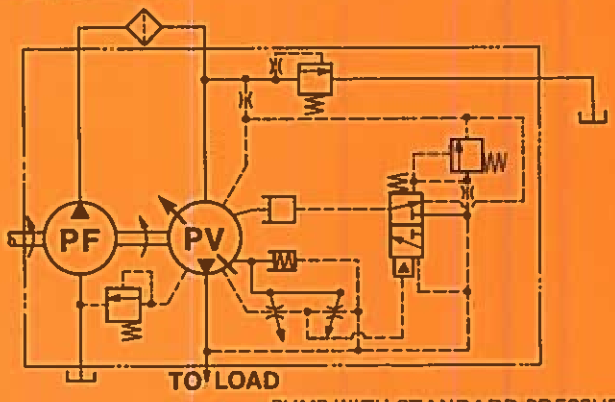


PUMP WITH STANDARD
PRESSURE COMPENSATOR

Torque Limiter

This control allows maximum performance to be obtained from the vehicle engine. It limits input torque into the pump to a preset value while the pressure compensator limits maximum pressure in the system.

When pump input speed remains constant, the torque limiter acts as an input horsepower limiter. At low load levels, the control permits high pump displacement, and thus high load speeds. With heavy loads, the load speed is reduced and engine stalling is prevented.



PUMP WITH STANDARD PRESSURE
COMPENSATOR AND TORQUE LIMITER

MODEL SERIES	DIMENSIONS IN INCHES* (MILLIMETERS)			SAE 4-BOLT PORT CONNECTIONS**					PILOT PRESS. CONTROL PORTS	DRAIN PORTS	APPROX. IN WTS. LBS. (Kgs)	DRIVE SHAFT SPLINE***
	L	W	H	PISTON PUMP INLET	PISTON PUMP OUTLET OR MOTOR INLET & OUTLET	CHARGE PUMP INLET	CHARGE PUMP OUTLET	CHARGE FLOW RETURN PORT				
PVD45	17.10 (434.3)	11.50 (292.1)	10.25 (260.4)	1.00 (25.4)	1.00 (25.4)	2.00 (50.8)	1.25 (31.8)	1.25 (31.8)	—	—	195 (88)	C
PVD80	20.40 (518.2)	12.60 (320.0)	12.44 (316.0)	1.25 (31.8)	1.25 (31.8)	2.00 (50.8)	1.25 (31.8)	1.50 (38.1)	—	—	342 (155)	E
PVD120	24.00 (609.6)	13.60 (345.4)	14.80 (375.9)	1.50 (38.1)	1.50 (38.1)	3.00 (76.2)	1.50 (38.1)	1.50 (38.1)	—	—	468 (212)	F
MFD35	10.53 (267.5)	7.32 (185.9)	7.19 (182.6)	—	1.00 (25.4)	—	—	—	—	1.062- 12UN-2B	63 (29)	C
MFD45	10.93 (277.6)	7.90 (200.7)	7.80 (198.1)	—	1.00 (25.4)	—	—	—	—	1.062- 12UN-2B	75 (34)	C
MFD80	13.08 (332.2)	8.54 (216.9)	9.55 (242.6)	—	1.25 (31.8)	—	—	—	—	1.312- 12UN-2B	158 (72)	E
MFD120	15.00 (381.0)	10.54 (267.7)	10.56 (268.2)	—	1.50 (38.1)	—	—	—	—	1.625- 12UN-2B	225 (102)	F
MVD45	14.30 (363.2)	10.33 (262.4)	10.00 (254.0)	—	1.00 (25.4)	—	—	—	.562-18 UNF-2A	1.062- 12UN-2B	135 (61)	C
MVD80	16.20 (411.5)	13.43 (341.1)	11.90 (302.3)	—	1.25 (31.8)	—	—	—	.562-18 UNF-2A	1.312- 12UN-2B	266 (121)	E
MVD120	18.50 (469.9)	14.80 (375.9)	13.60 (345.4)	—	1.50 (38.1)	—	—	—	.562-18 UNF-2A	1.625- 12UN-2B	435 (197)	F

* Motor dimensions shown are for side-ported units. For end-ported units and direct mounted POWER MATCH directional valves, consult your Vickers representative.

** Piston pump outlet, and motor inlet and outlet ports, are SAE 6000 psi (413 bar). All other ports are SAE 3000 psi (207 bar).

*** As recommended for rated pressure in accordance with SAE J744b. For straight keyed or other shaft ends, consult your Vickers representative.

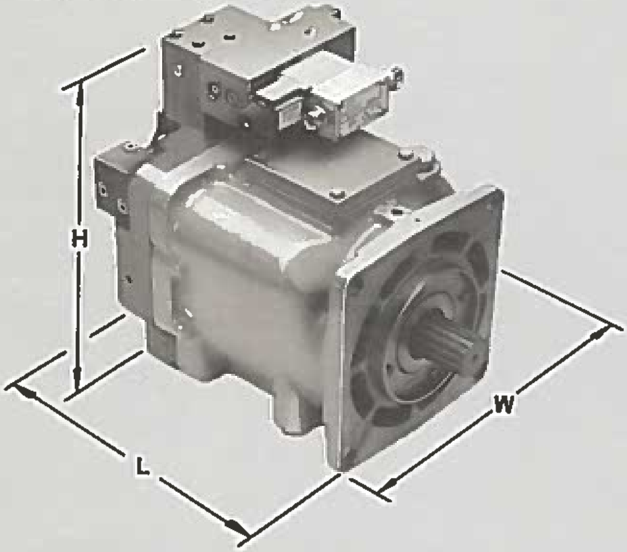
Mounting Flanges

All units have a 4-bolt flange mounting and pilot that conforms to SAE J744b "Hydraulic Power Pumps" standards. The 35 and 45 size units have "C" flanges; the 80 and 120 sizes have "E" flanges.

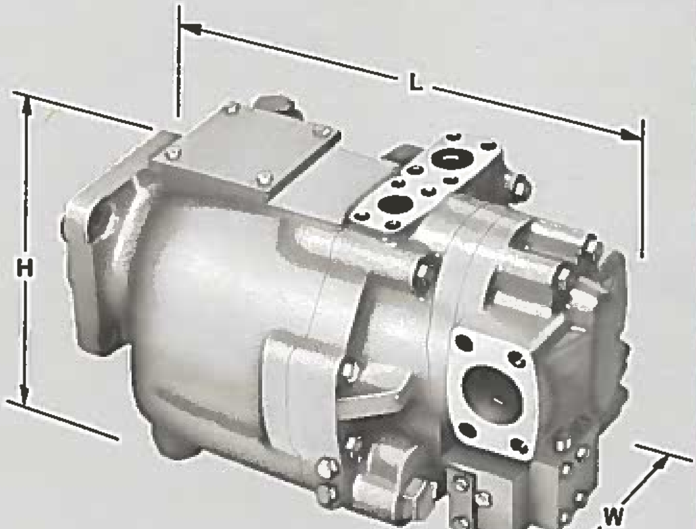
Shaft Extensions

Splined and parallel keyed extensions, in accordance with SAE J744b "Hydraulic Power Pumps" standards, are available.

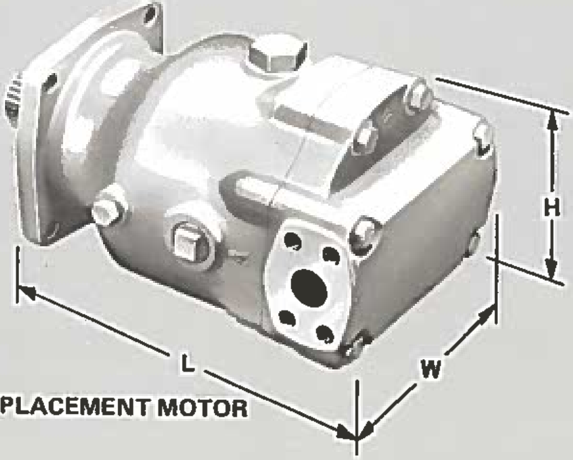
Dimensions



MVD VARIABLE DISPLACEMENT MOTOR



PVD VARIABLE DISPLACEMENT PUMP



MFD FIXED DISPLACEMENT MOTOR

Ratings/Application Data

PUMPS AND MOTORS ARE RATED AT 5000 PSI (350 bar) CONTINUOUS DUTY

PUMP SERIES	DISPLACEMENT IN.^{3}/REV. (CC/REV.)	SPEED – RPM*		THEORETICAL FLOW – GPM (L/MIN)		HORSEPOWER (kW)
		MIN.	RATED	@ 1800 RPM	@ RATED SPEED	
PVD45	5.77 (94.6)	500	2700	45 (170)	68 (257)	200 (149)
PVD80	10.28 (168.5)	500	2400	80 (303)	107 (405)	310 (231)
PVD120	15.40 (252.4)	500	2100	120 (454)	140 (530)	408 (304)

MOTOR SERIES DISPLACEMENT		DISPLACEMENT IN.^{3}/REV. (CC/REV.)	THEORETICAL TORQUE LB. IN./1000 PSI OR Nm (Kg/69 bar)	RATED SPEED* – RPM		THEORETICAL HORSEPOWER (kW)
FIXED	VARIABLE			FULL STROKE	7° STROKE†	
MFD35	—	4.49 (73.6)	714 (968)	2900	—	165 (123)
MFD45	MVD45	5.77 (94.6)	918 (1245)	2700	3300	200 (149)
MFD80	MVD80	10.28 (168.5)	1636 (2218)	2400	2900	310 (231)
MFD120	MVD120	15.40 (252.4)	2452 (3324)	2100	2500	408 (304)

* For speeds greater than rated, consult your Vickers, representative.

† Variable displacement motors

Charge pump inlet pressure range

Minimum . . . . 5 in. Hg Vacuum
Maximum . . . . 14.5 PSI (1 bar)

Maximum temperature, pumps & motors

Continuous . . . . 180°F (82°C)
Intermittent . . . . 220°F (104°C)

Typical efficiency at rated conditions

Overall . . . . Pumps 86%, Motors 87%
Volumetric (Pumps) 96%
Torque (Motors) 93%

Maximum motor case pressure . . . . 50 PSI (3 bar)
Minimum motor return line pressure . . 150 PSI (10 bar)

Rotation

The shaft rotation of motors is reversible. Shaft rotation of pumps is either clockwise or counterclockwise, as viewed from the shaft end.

Fluids

The fluid used should be in accordance with Vickers Fluid Recommendations Sheet M-2950-S.

Filtration

For satisfactory service life, provide full flow filtration to maintain total system cleanliness. Piston pump inlet oil should meet ISO code 18/13, which can be obtained using an inter-stage filter having 300 psi (21 bar) pressure integrity. Charge pump inlet oil should meet cleanliness as defined by ISO code 21/17. Consult your Vickers representative for filter selections from OFR and OFRS series.

Application assistance

POWER MATCH piston pumps and motors are designed to meet specifications as outlined. To assure maximum unit performance in conjunction with your specific application, consult your Vickers representative.

Model Code

Fixed Displacement Motors

MFD 35 - 1 - E - 20
1 2 3 4 5

1 Model Series

2 Nominal Displacement

@ 1800 RPM, 100 PSI (7 bar) in GPM (l/min)

35 (133)
45 (170)
80 (303)
120 (454)

3 Output Shaft

1 - SAE "C", MFD35
1 - SAE "C", MFD45
1 - SAE "E", MFD80
8 - SAE "F", MFD120

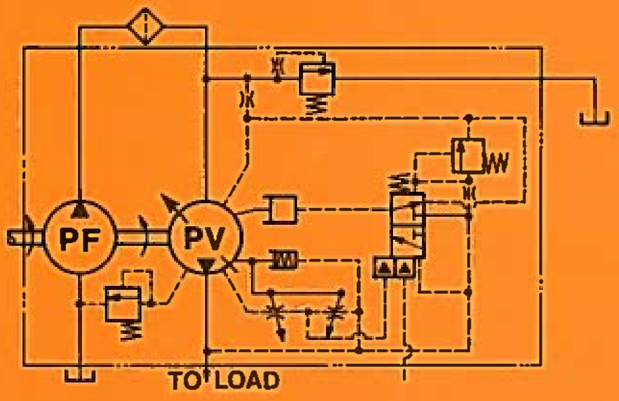
4 Port Options

Omit for side ports.
E - End ports
*E160 - Furnished with CMX160 valve
*E250 - Furnished with CMX250 valve
*E400 - Furnished with CMX400 valve
*Each CMX valve configuration must be specified separately.

5 Design Number

Torque Summer

On vehicles where one engine is driving two pumps, this control assures that pump demands do not exceed available engine torque. The control allows full engine power to be absorbed by either pump but limits the sum of the pump input torques to a level which will prevent engine stall.



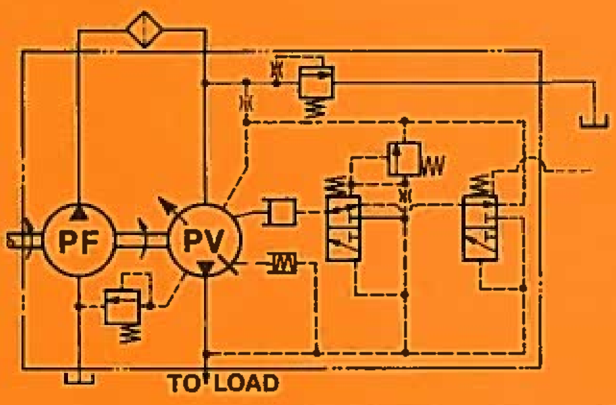
PUMP WITH STANDARD PRESSURE
COMPENSATOR AND TORQUE SUMMER

Load Sensing

This control provides a power match between the pump and load, thereby maximizing system efficiency and improving metering characteristics of the system directional valve.

The control senses pump outlet pressure and load pressure, and keeps the difference between these two pressures constant. The differential pressure is measured across a metering land in the system directional valve, with the effect that pump flow is determined by valve spool position and is independent of load pressure.

The load pressure signal must be generated in the directional valve, downstream of the metering land, and communicated to the pump's load sensor by a signal line. In a multiple-spool load sensing valve, the internal logic must sense all load pressure levels and feed the highest of these pressures back to the pump control.



PUMP WITH STANDARD PRESSURE
COMPENSATOR AND LOAD SENSOR

Load Sensing & Torque Limiter

This control combines features of the individual load sensing and torque limiting controls.

Load sensing and torque limiting functions are performed independently. However, when the preset input torque setting is reached, the torque limiter overrides the load sensor.

Load Sensing & Torque Summing

This control combines features of the individual load sensing and torque summing controls.

Dual-Pressure Compensator

This control allows two pressure settings, permitting the pump to supply oil to two functions operating at different pressures. For example, an excavator may require cylinder pressures limited to 4000 psi, but need 5000 psi for propel motors to provide adequate tractive effort. This is easily achieved with the dual-pressure compensator.

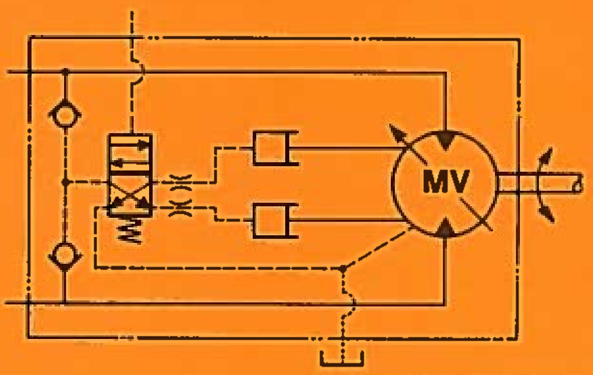
Electrohydraulic Control

Electrically modulated displacement control, which responds to a low power command signal, is available.

Motor Control

The control for variable displacement motors is a spring offset, spool type valve, either hydraulic pilot or solenoid actuated. The motors can be made available with the spring offset position giving either maximum or minimum displacement.

Applying minimum pilot pressure of 300 psi (21 bar) shifts the valve spool and directs outlet pressure to the control piston. The stroking rate is controlled by orifices in the control passages to the control pistons. Orifice sizes can be varied to control stroking time.



MVD TWO-POSITION MOTOR

Model Codes

Variable Displacement Motors

MVD 80 - 1 · 7 M E · 11
1 2 3 4 5 6 7

1 **Model Series**

2 **Nominal Displacement**
@ 1800 RPM, 100 PSI (7 bar) in GPM (l/min)

45 (170)
80 (303)
120 (454)

3 **Output Shaft**

1 - SAE "C", MVD45
1 - SAE "E", MVD80
8 - SAE "F", MVD120

4 **Minimum Displacement**

In degrees, 7° minimum

5 **Pilot Signal**

M - Pilot signal @ maximum displacement; no pilot signal, minimum displacement.
Omit if pilot signal @ minimum displacement; no pilot signal, maximum displacement.

6 **Port Options**

Omit for side ports
E - End ports
*E160 - Furnished with CMX160 valve
*E250 - Furnished with CMX250 valve
*E400 - Furnished with CMX400 valve
*Each CMX valve configuration must be specified separately.

7 **Design Number**

Variable Displacement Pumps

PVD 120 C L 1 - D - 12 - C 35 VT 28 - 14
1 2 3 4 5 6 7 8 9 10 11 12

1 **Model Series**

2 **Nominal Displacement**
in GPM (l/min) @ 1800 RPM, 100 PSI (7 bar)

45 (170)
80 (303)
120 (454)

3 **Charge Pump**

4 **Rotation**
(Viewed from shaft end)

R - Right hand (CW)
L - Left hand (CCW)

5 **Input Shaft**

1 - PVD35/45 (SAE "C", 14T, 12/24 DP)
1 - PVD80 (SAE "E", 13T, 8/16 DP)
8 - PVD120 (SAE "F", 15T, 8/16 DP)

6 **Shaft Seal**

S - Single D - Double

7 **Pump Design Number**

8 **Compensator Type**

C - Standard pressure compensator
CD - Dual pressure compensator

9 **Pressure Compensator Setting**

in tens of bar, 170-350 range (equivalent to 2000-5000 PSI); 35 = 350 bar

10 **Optional Controls**

(To be combined with standard "C" compensator; omit if not required)

T - Pressure limiter with torque control
TS - Pressure limiter with torque & summation controls
V - Pressure limiter with load sensing
VT - Pressure limiter with load sensing and torque limiter
VTS - Pressure limiter with load sensing, torque control and summation control

11 **Pressure Setting of Torque Controls**

In tens of bar, 20%80% of maximum pressure limiter setting; 28 = 280 bar

12 **Control Design Number**

VICKERS
1401 Crooks Road – Troy, Michigan 48084
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